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PRODUCT ENGINEERING

APRIL
1935

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Hans Ernst, outstanding designer of hydraulic systems for machine operation, explains how flow resistances in hydraulic circuits can be used to actuate valves to gi ve automatic sequential machine operation . . . . . PAGE 122

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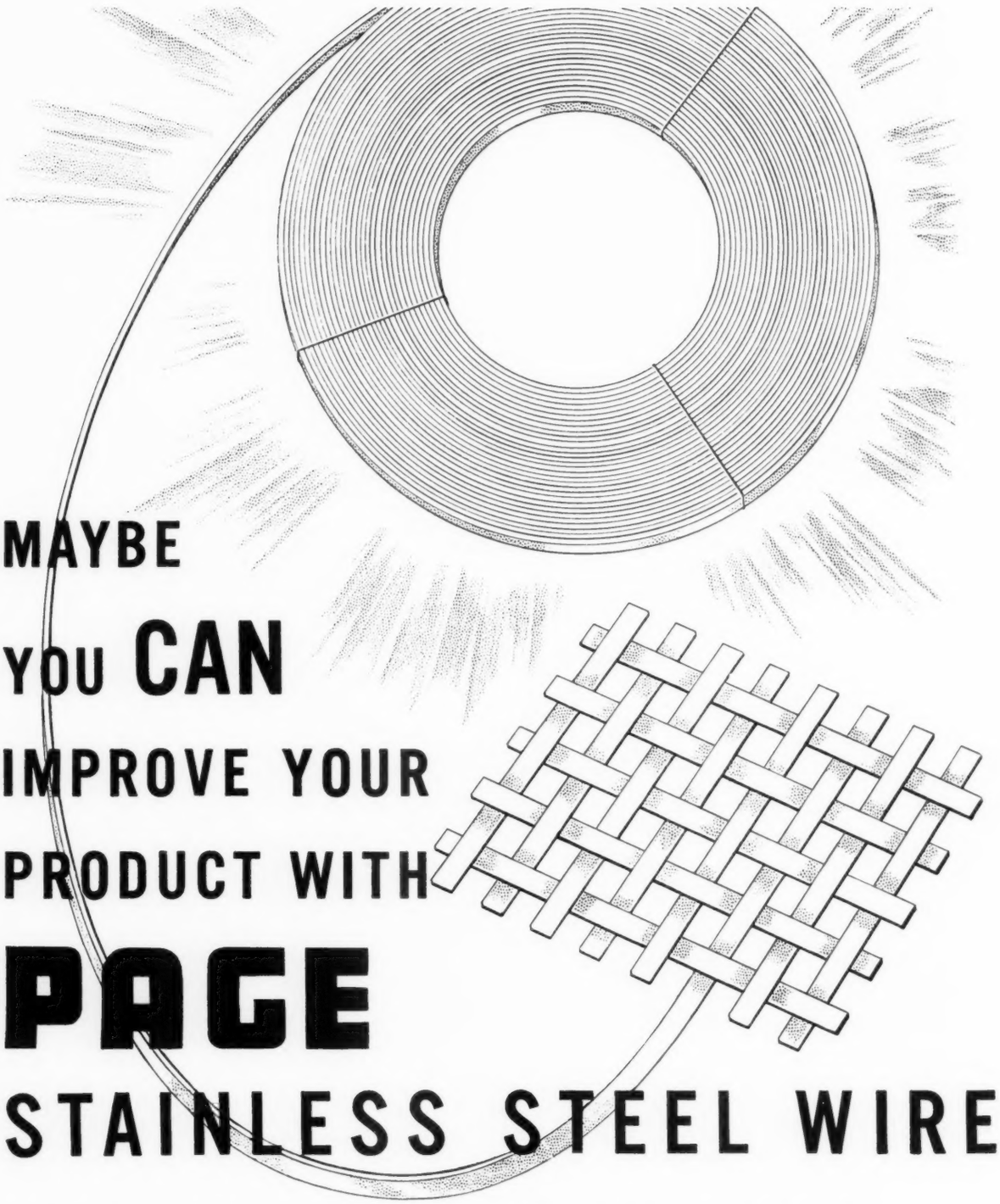
Mechanical Design • Product Development

McGRAW-HILL PUBLISHING COMPANY, INC.



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PRODUCT ENGINEERING

VOLUME 6 NUMBER 4

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Next Month

Included in May *Product Engineering* will be two unusual articles that will have special appeal:

First, the second installment of Hans Ernst's series on hydraulic circuits, wherein he shows the practical application of hydraulic choke coils and resistances for controlling sequential operations in machines.

Second, an analysis of the factors that enter into the selection of chain drives. This article will show how to evaluate the advantages and disadvantages of finer pitches, greater numbers of sprocket teeth, chain lubrication and other considerations, as affected by speed, load and service conditions.

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Intimate Correspondence

In deference to the expressed wishes of some of our readers, letters published on this page will give only the initials of the writers unless specific permission is given to use the name.

Not Enough Details

PRODUCT ENGINEERING
NEW YORK

What is the idea of giving an interesting article such as "Developments That Made the Motor Cars of 1935," but then leaving us high and dry when it comes to telling how to duplicate the things that automobile designers are doing? For example, in March this article stated that chain timing drives were getting more popular, thermostats for automatic controls are being used more than ever, live rubber is used for mountings and that prelubricated ball bearings are in general use.

All this makes interesting reading but it is not much help to me. Why not give details about the design of thermostats and controls as used in autos, types of rubber mountings, ball bearing housings, etc.?

E.R.M.
Providence, R. I.

EVIDENTLY E. R. M. wants both a survey of the developments taking place and details of design all in one issue. We judge that our automobile articles hit the mark with E. R. M. in so far as interest is concerned, and as to "design helps," we feel sure that our articles on thermostat applications and chain drives next month will be gratifying to E. R. M. We wonder how some of the other readers feel about it.

We Want Color!

PRODUCT ENGINEERING
NEW YORK

In answer to the letter from H. E. Taylor as printed in the March number, you stated "we are now negotiating . . . for a series of articles on color." Could you tell me when they will begin and who is the "recognized authority?" I agree with Mr. Taylor in wanting at least something on color and finishing.

B.H.H.
Pittsburgh, Pa.

THAT makes it unanimous as far as we are concerned. We cannot commit ourselves on how soon the author will have his articles ready for us, but we can let you in on who the "recognized authority" is. He is no other than Mr. Howard Ketcham, editor of the "Automobile Color Index," published by Du Pont, for eight years director of the Duco Color Advisory Service of E. I. du Pont de Nemours & Company and now color consultant for a number of the leading automobile makers, several manufacturers of business and household equipment and numerous other companies.

Maybe We Slipped

PRODUCT ENGINEERING
NEW YORK

Usually the way you set up your engineering data sheets it makes them easy to use. But you sure slipped on the "Spring Data" in the March issue, I almost got dizzy picking out all the items that applied to Type 2 springs.

E.H.W.
New York, N. Y.

PERHAPS the data referred to by E. H. W. could have been presented in a better manner, but there is always the problem of space limitations. We can only plead that we arranged the data to fit and tried to make it as easy to read as possible. Evidently we did not do well enough to suit E. H. W. and we admit every job can always be done better. We'll give it more thought next time we have a similar article.

Top Hole

PRODUCT ENGINEERING
NEW YORK

It certainly is pleasing, and at the same time good for engineers, to find a section called "Question and Comment." I would refer specifically to the articles in the January number by Mr. J. Struthers Dunn and Mr. Walter Teague's modest article. To me it is the first interesting section.

C.A.U.
Waltham, Mass.

LOTS of readers get the "feel" of the profession by looking first at the questions and answers. We do ourselves.

More Applied Mechanics

PRODUCT ENGINEERING
NEW YORK

Received my *Product Engineering* a few days ago and must say that I am well satisfied with it.

The article "Lateral Vibration of Shafts" is, I think excellent. Since it is the view of a fellow worker in the field, it gives you a good check on your own ideas. I have this problem to solve many times in my work. This article appealed to me because it gave a solution to a complex problem in a straightforward way; and there are many involved problems that can be handled in like manner.

Please give us at least one good article on applied mechanics each month.

Has anything appeared in the past on high speed bearings for steam turbines? Babbitt lined shells 4000 to 6000 r.p.m. I am interested in the design of such bearings and their supports. You would favor me with a reply on this matter.

J.H.T.
Hyde Park, Mass.

YOU can rest assured that we will do what we can to give you one good article on applied mechanics in every number of *Product Engineering*. Of course we feel that merely to give a purely theoretical article does not meet the needs of our readers. Therefore, we confine ourselves to such articles on applied mechanics that translate the theories into practical working formulas.

With reference to articles on high-speed bearings, there was an excellent article on this subject in the January, 1933, number of *Product Engineering* wherein Mr. Schein of the Sperry Gyroscope Company discussed the various stages in the development from a common sleeve type bearing to the highest type of sleeve type bearing such as used in gyroscopes.

Who Are the Manufacturers?

PRODUCT ENGINEERING
NEW YORK

In the December, 1934, issue of the *Product Engineering*, the article on "Needle Bearings and Their Application," page 461, we are interested in knowing the name of the one manufacturer who is furnishing needle bearings with all parts made of stainless steel. We would appreciate your sending us this information. Why don't you publish the information in the article?

L.H.
Beloit, Wis.

ALTHOUGH we have no objection to publishing the names of manufacturers of a type of product that is not generally available, it would be impractical for us to attempt to list in our articles the names of manufacturers of parts and materials that are made almost universally.

With reference to manufacturers who furnish needle bearings with all parts made of stainless steel, we know that at least three of the leading manufacturers of needle bearings can fill this specification. Perhaps if we inquired of all of the other manufacturers we would find that they also are in a position to furnish stainless steel needle bearings.

Incidentally, this question has come up repeatedly and we wonder what our readers think relative to the idea of publishing manufacturers lists with the articles.

PRODUCT ENGINEERING

GEORGE F. NORDENHOLT, *Editor*

MCGRAW-HILL PUBLISHING CO., INC. APRIL 1935 VOLUME SIX, NUMBER FOUR

The Competition for Leadership

IN a recently released publicity item there appears the statement, "The United States is supreme in design and development of machine tools and can rely on remaining in the leading world position for at least another twenty years."

Every industrial nation has been watching closely American developments in machine tool design and construction. Not only have foreign nations been quick to adopt American improvements in machine tools, but they have at the same time developed other improvements of their own. One English machine tool manufacturer has concentrated his efforts on lighting and optical equipment as an integral part of his machine. From Germany come reports of striking developments in electric drives and controls. And a study of the designs of many foreign machines will reveal other improvements.

There is no lack of confidence in the initiative and ability of American machine tool designers. Because of the great diversity of production equipment required and the nature of our manufacturing methods, there is a continuous demand for better manufacturing tools and equipment, thereby giving American machine tool designers many opportunities for developing improved products that will be marketable.

But with Russia making every effort to develop her own machine tool industry, and Germany and England striving to capture foreign markets, it will need more than mere faith for America to retain her leadership. It will require that the intensive development programs of the past five years will have to be continued with unabated activity.

Although the above refers specifically to the machine tool industry, it applies equally well to other fields of product design. The rapid advances in Great Britain and Germany and the industrialization of Russia require American manufacturers to make every effort to retain their leadership.

There is no question as to our supply of technological skill of the highest caliber. And backed by the scientific and industrial research laboratories American designers have available an abundance of material with which to achieve the desired results.

BUT nothing can be accomplished in new developments unless the management of the manufacturing company takes a progressive attitude. The engineering departments must be properly manned. Appropriations must be made for the necessary development expenses. Extensive laboratory tests must be undertaken to verify the engineering calculation or uncover deficiencies. Proper manufacturing equipment must be procured, and an aggressive advertising and marketing program must be launched.

Companies sometimes find themselves so situated that conditions do not appear to be favorable for continued development activities, but any laxation is likely to be fatal. Those companies or industries that, yielding to the pressure of circumstances, cease their development activities, usually soon join the ranks of those that are forgotten.

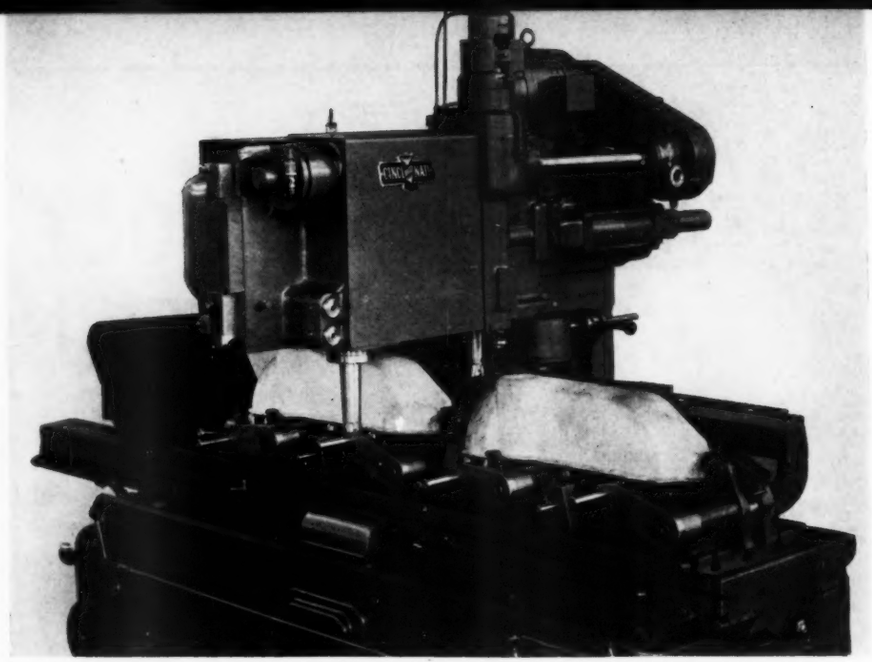


Fig. 1 — Cincinnati Hydromatic milling machine with special control circuit. A tracer arm contacting a profile cam causes the cutters to follow a predetermined, irregular path around the work

Modern Hydraulic Control Systems and Circuits—I

HANS ERNST
Research Engineer
Cincinnati Milling Machine & Cincinnati Grinders, Inc.

DURING the past decade the industrial world has seen a remarkable expansion in the field of hydraulic power transmission.

This is all the more striking in view of the fact that the application of hydraulic power to large presses, elevators, and heavy machinery was quite well developed more than 40 years ago; in fact, a central power station for the distribution of hydraulic power for such purposes was installed in London, England, in 1883. Not until recent years, however, has any real headway been made in the application of hydraulic power in the field of machine tools and other small automatic machinery.

The hydraulic renaissance which is now so evident throughout the world had its inception in three important developments:

1. Improvements in the design of small, self-contained, high-pressure pumps, of both the variable and the constant displacement types.

2. Fuller recognition of the pernicious effects of entrained air on the motion of hydraulically propelled slides, and improvement in means for its elimination.

3. The development of the "hydraulic circuit" idea, closely analogous, in fact, to the electric circuit, and the design of particular circuits with characteristics suited to the requirements of each individual application.

In the old days, with accumulator systems, there were no elaborate hydraulic circuits comparable to electric circuits—there was simply the pump, the pressure tank or accumulator, and supply lines leading to the operating cylinders. Simple? Perhaps so, but also with all the faults and crudities that go with simplicity. Modern hydraulic circuits on the other hand, are often highly complex, a variety of special devices, such as automatic valves, hydraulic choke coils, and other elements being arranged in proper sequence to provide the desired operating characteristics.

A typical example of a machine for which a relatively complex hydraulic circuit is required is illustrated in Fig. 1. In this machine two milling cutters are arranged to follow a predetermined irregular path around a work piece, as indicated in Fig. 2 by the dotted line.

In the electrical field, development has followed a double path—unit development and circuit development—and the functioning of an electrical system depends as much on the particular arrangement of the circuit as on the detail construction of the associated units. In the latest developments in the electrical field, namely, radio engineering, the importance of the circuit is still more evident.

And so with hydraulics—recent developments are largely circuit developments, and in the modern hydraulic circuit there is an extremely close parallel to the electrical circuit. A slight change in circuit—with the same elements—may produce entirely different operat-

Hans Ernst, who developed the "hydraulic-electric analog" method of designing hydraulic circuits, as described here, is recognized as one of the foremost authorities on the design of hydraulic systems. The circuits shown diagrammatically in this article are taken from patents owned by Cincinnati Milling Machine Co.

ing characteristics, as will be explained in these articles.

Considerable information has been published in recent years regarding the various types of pumps available for use on hydraulic systems, hence this phase of the subject need only be briefly referred to here. A study of the Patent Office files and technical journals will reveal hundreds of different forms of hydraulic pumps. Nevertheless, most of these forms, particularly those in commercial use, may be classified under three headings: plunger type, vane type and gear type. Typical examples of gear and vane type pumps are shown in Fig. 3, and in Fig. 4 is shown, in section, a plunger type pump.

In the majority of designs the plunger type pump is made with variable displacement while the vane and gear pumps are made with constant displacement—though variable displacement vane pumps are also used in certain applications. In each particular application, the type of pump to be used is a matter for careful consideration.

Where the requirements of the system are such that hydraulic slip or leakage must be kept to a minimum, where the unit pressure required is relatively high, or where the rate of motion of the driven member must be varied within wide limits with a minimum power loss, the variable displacement plunger type pump is generally used. Sometimes, however, the advantage of lower cost in the constant displacement vane or gear pump may outweigh the disadvantage of higher power loss. On the other hand, where the total hydraulic power involved is not excessive, where exact maintenance of a set feed rate is not so important, or where the rate of motion does not vary over wide limits, the conditions favor the use of the vane and gear pumps. Furthermore, by the use of special control valves and compensating devices it is now possible to maintain set feed rates to quite close limits, even with constant displacement pumps, hence the field for these pumps has been greatly extended in recent years. However, where an exactly measured flow is required, the choice still lies in favor of the high-grade plunger pump.

Complexity in modern hydraulic circuits is chiefly the result of an increasing complexity in the functions of the machines to which hydraulic systems are being

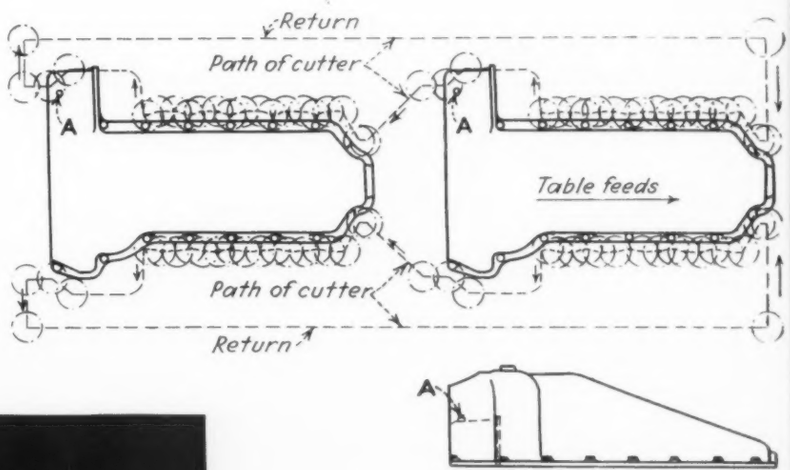


Fig. 2 — Diagram showing path of cutters in going around the work in the hydraulically controlled milling machine illustrated in Fig. 1

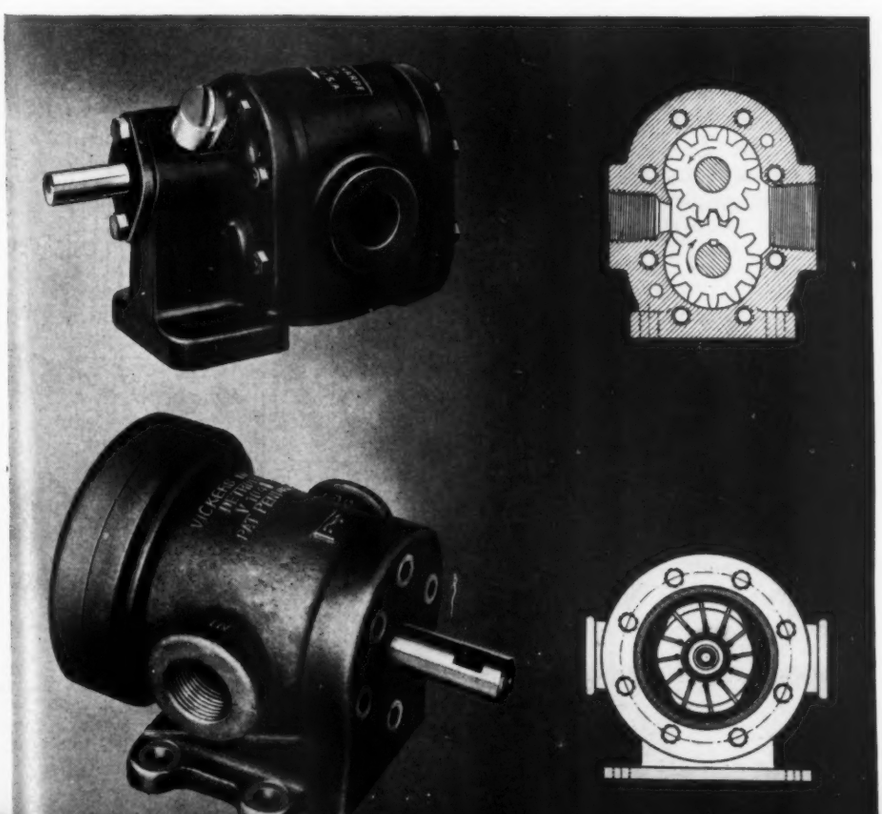


Fig. 3 — Typical gear type Brown and Sharpe constant displacement hydraulic pump and, below, a Vickers variable displacement vane type pump

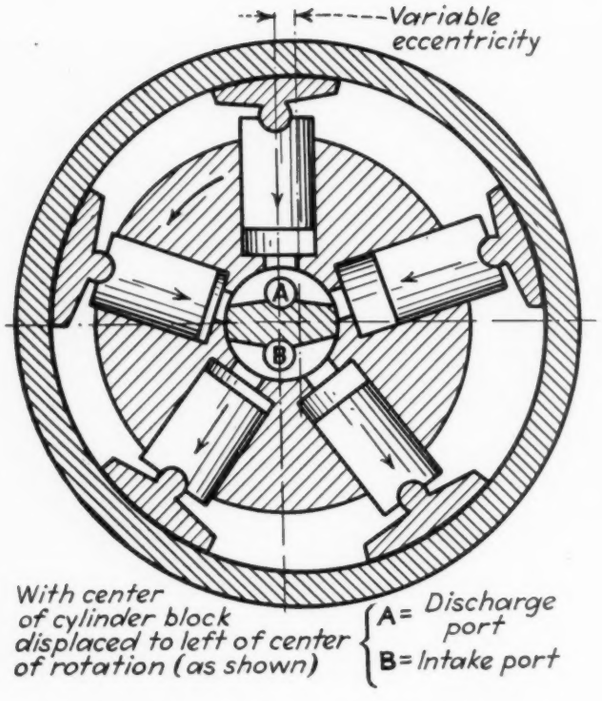

Variable eccentricity
A
B
With center of cylinder block displaced to left of center of rotation (as shown)
A = Discharge port
B = Intake port

Fig. 4—Diagram showing principle of operation of typical plunger type pump with variable displacement

applied. In most of the early machine applications, the hydraulic system was called upon to produce motion in a single reciprocating member only, whereas in the modern machine the hydraulic system may be required to operate a multiplicity of slides—all in properly timed sequential relationship—together with a variety of interlocks, clamping devices, and similar elements.

It is evident that much care must be given to the design of the circuit in order to obtain the required functions with the minimum outlay and minimum power consumption. To provide a separate pump for each simultaneous function would often entail too great an initial expense, hence it is usual to distribute the flow from a single pump to several independently operating members, there to be used in accordance with requirements. In many applications, a variable displacement pump is used for the main operating function of the machine and a constant displacement pump for the various auxiliary functions.

In the division of flow from a common source to several operating branches, many difficult problems are encountered, especially where the rate of flow in any particular branch must be closely controlled. In fact it may rightly be said that the solution of problems associated with the division of flow forms the basis of many recent hydraulic developments.

Types of Fluid Flow

Before entering into a discussion of various circuits and control mechanisms, we must consider some fundamental facts associated with the flow of fluids. In the past the emphasis has been placed too much on the hydraulic head, or pressure, existing under a given set of conditions. From our conventional viewpoint we have considered a pump primarily as a pressure-producing mechanism instead of a means for conveying fluid from one place to another, the pressure developed being merely incidental to the opposition to flow which exists in the discharge line. This viewpoint has arisen quite naturally from the historical development of hydraulics.

In the earliest applications of hydraulic power, the source of fluid energy was usually a body of water existing at an elevation higher than the point of utilization. Thus, the one fixed fact known was the available head. The pressure drop in any pipe, or energy transforming device connected to this source, was merely that produced by the difference of elevation between the surface of the fluid in the reservoir and the point of release to atmosphere. Consequently, the resistance to flow in a pipe line has sometimes been expressed as the pressure drop over the total length of the pipe multiplied by the cross-sectional area—that is, as the total force acting on the column of fluid, or the total opposition to flow in the line.

It is a well-known fact that the motion of a fluid in a pipe may be of two kinds. In the one type there is a steady, continuous motion of the particles parallel to the axis of the pipe, generally termed streamline, viscous or laminar flow. In the other case, the motion is not steady, but turbulent, being accompanied by the formation of eddies. This type of motion is variously described as turbulent, unsteady, or sinuous flow. Under any given set of conditions the motion may be streamline at low velocities and turbulent at high velocities, the transition from streamline to turbulent flow taking place within a very definite range of velocities determined by such factors as the diameter of the pipe and the viscosity and density of the fluid. Where the diameter of the pipe is small and the viscosity of the

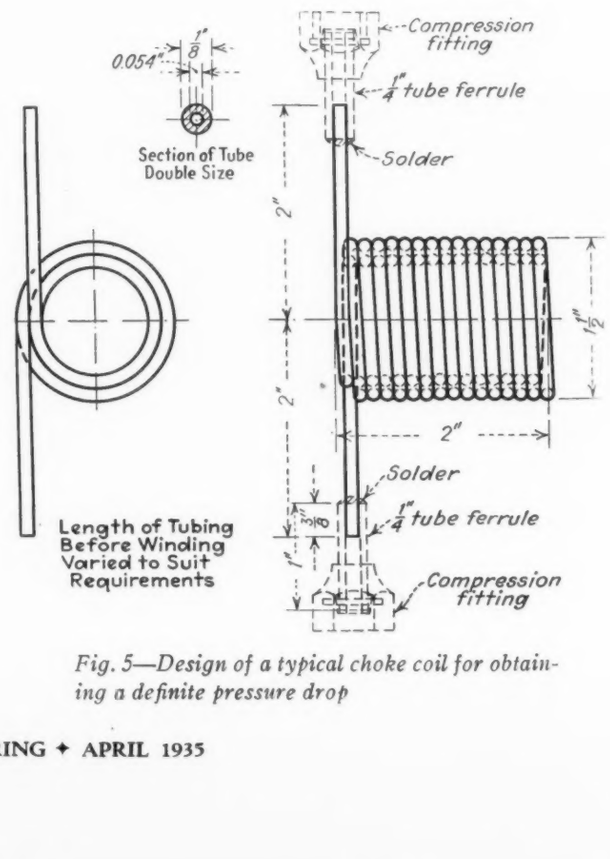


Fig. 5—Design of a typical choke coil for obtaining a definite pressure drop

fluid is high, as in the case of oil, and where the quantity or rate of flow is relatively small, the motion is definitely streamline.

In the application of hydraulic power to machine tool feeding mechanisms and the like, streamline flow is frequently encountered, particularly when dealing with relatively low feed rates. In choke coils or bleeders formed of small bore tubing, as shown in Fig. 5, and in all cases of hydraulic leakage or slip, the flow is streamline.

Whenever streamline flow is encountered, the pressure drop produced in a given pipe is directly proportional to the velocity. This is usually expressed by the following equation, established by Poiseuille:

$$P = \frac{32\,\mu\,L\,v}{d^2}$$

P = pressure drop in lb. per sq.ft.
μ = absolute viscosity of the fluid in poises
L = length of the pipe in ft.
d = diameter of the pipe in ft.
v = average velocity of fluid through pipe in ft. per sec.

But the average velocity of the fluid is equal to the quantity Q, divided by the area, $\pi d^2/4$. Hence we may rewrite the above:

$$P = \frac{128\,\mu\,L\,Q}{d^4}$$

If P is to be expressed in 1 lb. per sq.in., L in in., d in in. and Q in cu.in. per min., then:

$$P = 0.0047\,\frac{\mu\,L\,Q}{d^4}$$

If we now divide through by Q, we obtain the relation:

$$\frac{P}{Q} = \frac{0.0047\,\mu\,L}{d^4}$$

On the left-hand side of the equation we now have the pressure drop per unit of flow, while on the right-hand side we have a quantity which depends only on the physical dimensions of the pipe and the viscosity of the fluid which passes through it. This is therefore an inherent characteristic of a particular pipe with a particular fluid and thus constitutes a true resistance. Designating this by R we may write $P/Q = R$, and thus establish a value of R in terms of units derived from the customary pressure and quantity units, or again:

$$P = Q\,R$$
Pressure = Quantity × Resistance

This is directly analogous to the well-known Ohm's Law of the electrical circuit, $E = I\,R$. Of course, the objection will immediately be raised that the value of R in the equation $P = Q\,R$, is not constant but will vary with every change in the viscosity of the fluid; furthermore, that even for any given viscosity, R will be constant only with perfect streamline flow. This is true, but the fact that R may vary with a change in conditions does not nullify the validity of the statement $P = Q\,R$, where R has the value determined by the conditions at the instant under consideration. In like manner, in the electrical equation $E = I\,R$, the value of R is not always constant, but varies with the temperature, frequency, etc. But this does not prevent its application in the solution of problems where these conditions are assumed to be constant, nor does it detract from its usefulness in the analysis of complex circuits.

Resistance As An Entity

The chief usefulness of the expression $P = Q\,R$ is that it establishes the resistance R as an entity—an independent variable determined by a particular circuit condition. The pressure drop P now becomes the dependent variable whose value is the result of forcing the quantity Q through this particular resistance. Thus, when unit flow produces unit pressure drop, we are encountering unit resistance.

In practice, definite pressure drops in a hydraulic circuit are obtained by choke coils and bleeders formed of small bore tubing. These choke coils are usually made of $\frac{1}{8}$ in. O.D. copper tubing of 0.054 in. bore, and length ranging from a few inches to more than 30 ft., depending on requirements. For convenience, the chokes are

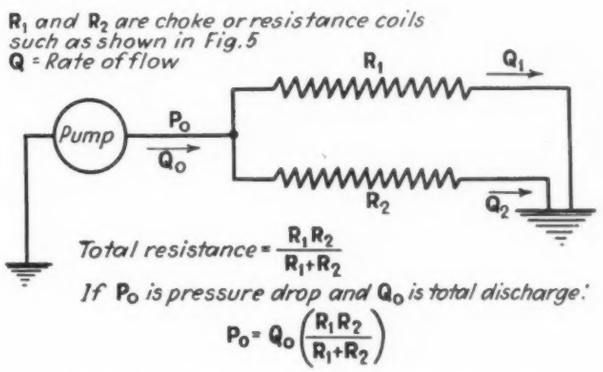

R1 and R2 are choke or resistance coils such as shown in Fig. 5
Q = Rate of flow
R1
Q1
Pump
Po
Qo
R2
Q2
Total resistance = R1R2 / R1+R2
If Po is pressure drop and Qo is total discharge:
Po = Qo (R1R2 / R1+R2)

Fig. 6—Illustrating the relation of pressure, flow and resistance when hydraulic resistances are connected in parallel

usually wound in the form of a helical coil of one or more layers. The pressure drop across the choke or resistance coil may be utilized in a hydraulic circuit to operate either pilot valves or control valves, just as the voltage drop across an electrical resistance is used to send current through a circuit shunted around it.

From the above viewpoint it is clear that we can arrange and combine hydraulic resistances in much the same manner as electrical resistances, and thus obtain similar effects. Combinations of hydraulic resistances in series and parallel may be analyzed in precisely the same manner as similar combinations of electrical resistances. Thus, in Fig. 6 the total resistance of R_1 and R_2, connected in parallel, is equal to the reciprocal of the sum of the reciprocals of the individual resistances, or $R_1\,R_2$ divided by $(R_1 + R_2)$, while the division of flow will be in accordance with the ratio of the individual conductances $1/R_1$ and $1/R_2$ to the total conductance $R_1 + R_2/R_1\,R_2$.

Just as in electrical circuits, numerical values for hydraulic resistance can be established, and the resistance required for a desired pressure drop thereby calculated. This opens an almost unlimited field of possibilities for new devices and circuit arrangements, to be explained in the next article of this series.

PRODUCT DEVELOPMENTS

• *Greater Rigidity for Roughing Pinions*

To obtain high accuracy in the final machining of spiral-bevel and hypoid pinions, even the roughing cuts must be made accurately. But the greater amount of metal removed requires greater rigidity, greater power and better lubrication than needed for lighter and more accurate final machining. In the design of the machine shown here, built by the Gleason Works, rigidity throughout the design of the machine has been uppermost in the minds of the designing engineers. But to take full advantage of production possibilities accompanying this increased rigidity, the adoption of hydraulic drives, automatic operation, and built-in lubrication not only allows a higher production rate, but also increases tool life.

Many manual functions that formerly required the attention of the operator are now accomplished automatically through hydraulic control valves. Work is clamped in the chuck by a hydraulic unit, a releasing device operating automatically at the end of the cut. Circular movements of the cradle and work spindle for generating the tooth shape are derived from reciprocating motions of a pair of racks actuated by a hydraulic piston. A hydraulic brake takes up back lash in the generating train. In-and-out feed of the work head to permit indexing is also hydraulic, while the rate of feed and the change from rapid in-feed to a slower cutting feed is under hydraulic control.

To assure rigidity, the machine frame is cast in one piece with the cutter cradle and the work head mounted on separate columns tied together at the top.

Spindle drive is from a separate motor through three gears, including speed change gears. The hydraulic pump and coolant pump are driven by a second motor with all electrical controls built into the machine. A "plugging" switch stops the spindle motor from coasting. This motor starts automatically when a manually-operated clamp tightens the work in place.

In addition to automatic control of the motor drive and the chucking and feeding operations, lubrication is entirely automatic for all bearings in the machine. Satisfactory performance of the entire unit at its greatest capacity, therefore, does not depend upon periodic attention to the lubrication of a large number of bearings.

With increased rigidity in the Gleason pinion rougher, more accurate roughing cuts can be made. Automatic controls and built-in lubrication reduce need of attention from the operator

• *Dredge Bucket Arc-Welded*

A weight reduction of 50 per cent was not the only advantage obtained in replacing cast steel dredge buckets with buckets of arc-welded design shown here. The Argentine Government Shipyards, where the welded bucket is used, reports increased performance, reduced power requirements and less wear on the dredging machinery. According to the Lincoln Electric Co., the arc-welded design weighs but 1,300 lb., whereas the cast steel bucket of equal capacity weighed 2,600 lb.

This arc-welded dredge bucket weighs but half as much as the cast steel bucket formerly used

The body of the welded bucket is made of sheets ½ in. thick, with the frame and cutting lip formed from 1-in. sheets. The corner-drilled pieces in the bottom are 2½ in. thick, and were cut from heavy plates by an automatic gas cutting machine.

• *Hydraulic Drive for Home Appliances*

Smooth cushioned return in the operation of a new flat plate ironer is assured, in addition to automatic control of closing and the pressing force, by the use of a Thrustor, a motor-driven hydraulic device built by General Electric. Although this hydraulic unit has been widely used in industry, this is its first application to home appliances.

By controlling the motor-driven pump in the hydraulic unit by pressure on the control handle, the levers and button controls formerly used were eliminated entirely. In addition to simplifying the operation of the ironer the straight line motion given by the hydraulic unit eliminates the necessity for the cams, springs, gears and racks ordinarily used.

Vibrations set up by electromagnets feed material from the hopper to the discharge end of the trough in the Syntron feeder

• *Optical Equipment for Jig Bores*

In the new Société Genevoise jig borer, increased power and rigidity built into the tool permits heavy duty work without sacrificing accuracy. As the result of the development of a new precision hydraulic drive combined with an optical system of locating work the new jig borer can also be used as a production machine tool.

For accurate setting of the table, a hydraulic traverse provides a range of feed from 0.05 to 80 in. per min. under control of the hand lever shown at the right in the accompanying illustration. When the hand lever is released, it returns to the center position and locks the table in place. For milling, the lever can be locked into any position.

Readings for the position of both the table and the spindle saddle are taken from the exposed scales appearing in the illustration or from precision scales mounted inside the machine and viewed through microscopes. Graduations on the precision scales are spaced 0.1 in. apart and are engraved on a polished surface on the inside of a channel section. In the design of the scale for the table the sides of the channel are closed against the frame by spring pressure, with the joint lubricated from the guide way by oil fed through a felt filter.

Prismatic microscopes built into the bed and crossrail are focused on the scales and adjustable for a short distance along the scale by a micrometer screw readable to 0.0001 in. The table position is given by the nearest graduation on the scale plus the amount of adjustment required to center the microscope over the graduation. Light for reading the scale is supplied by a 4.5-volt bulb built into the microscope housing. When not in use, the microscope shown in the illustration is swung to the right and inclosed by a protecting cap.

In the new Société Genevoise jig borer the table position is determined from precision scales viewed through microscopes

• *Electromagnet Drive for Vibratory Feeder*

The use of electromagnets for vibrating dry feeders not only reduces the number of wearing parts, but also simplifies control over the rate of feed. An example is the design shown here developed by the Syntron Company. An electromagnet drive permits adjustable feeding capacity from 2 oz. to 2,000 lb. per hr., with the complete elimination of such wearing parts as motor shafts, gears, bearings, connecting rods and slide valves. In addition, the rate of feed is instantly controlled by the rheostat mounted on the panel. With the rheostat once set for a given feed, the rate of flow is said to remain constant.

Operating on alternating current, a vibrator under the trough sets up vibrations at the rate of 120 per

sec. The electromagnet rapidly draws the trough down and toward the hopper against a flat spring. The downward motion is so fast that material in the trough remains stationary. The slower forward and upward return movement under the action of the spring alone lifts the material and carries it toward the discharge end.

Volume of flow is regulated by controlling the current through the feeder vibrator, thus changing the length of the vibrator stroke. An ammeter on the panel indicates the amount of the vibrator current, and therefore the volume of flow.

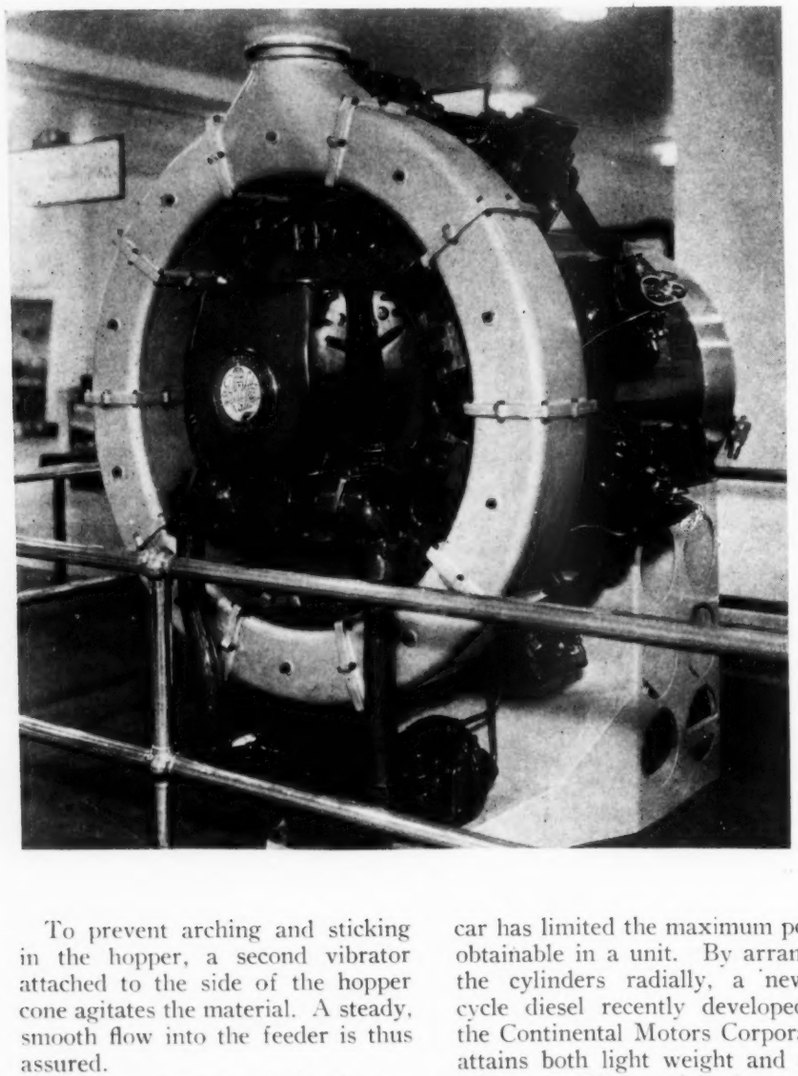
To prevent arching and sticking in the hopper, a second vibrator attached to the side of the hopper cone agitates the material. A steady, smooth flow into the feeder is thus assured.

Free movement of the hopper within a restricted zone is permitted by shear rubber mountings at the points of support. Rubber mountings of a similar type under the frame of the machine prevent transmission of vibrations to the floor or machine support.

• *Light-Weight Diesel for Rail Cars*

To reduce the weight per horsepower has been one of the major problems in the design of diesel engines for rail cars. Also, the limited space available in the conventional car has limited the maximum power obtainable in a unit. By arranging the cylinders radially, a new 2-cycle diesel recently developed by the Continental Motors Corporation attains both light weight and compactness. With a 6½-in. bore and a stroke of 7½ in., the ten-cylinder design shown in the illustration develops 635 hp. at 1,400 r.p.m. and weighs only 3,900 lb. This weight, which includes accessories, gives a ratio of only 6 lb. per hp. With a direct current generator of special design attached to the engine, the entire unit weighs 9,200 lb., or about one-third the weight of the conventional diesel-electric unit of equal power for rail car service. The entire radial unit is only 71 in. in diameter and 82 in. long, including the generator.

Radial diesel engine designed for rail car service develops 635 hp. for a weight of 3,900 lb. Including a direct-connected generator, the power plant is less than 7 ft. long



Light weight and compactness is obtained principally by the radial construction, which allows a large reduction in the weight of the crankcase and crankshaft as well as reducing the length. As in aircraft engine practice, one master connecting rod carries nine articulated rods. Two roller bearings and one ball thrust bearing support the short, two-piece crankshaft. Wrist pins are carried on needle bearings.

Scavenging air is admitted through sleeve valves which form the cylinder bore. Except for a short interval near each end of the stroke, the sleeve valve moves in the same direction as the piston, reducing the rubbing speed of the piston to about 900 ft. per min. Uniflow scavenging is also obtained, air entering the cylinder through ports uncovered by the sleeve at the end of the stroke and leaving through ports near the top during the compression stroke.

Reduction in the weight of the crankcase is the result of extensive use of high-strength alloy steels and welded construction rather than the selection of light-weight alloys. Steel castings are used for only a few small parts which could not be readily welded.

• *Cork and Copper Seal*

To protect the final drive in Caterpillar tractors, each side of the sprocket hub is sealed with an oil-retaining, dust-excluding cork and copper bellows seal. Within the seal, shown in the illustration, oil lubricates the rubbing surfaces and works outward, carrying dirt away from the outer edges and preventing the entrance of water.

Cork sealing surfaces are pressed against corrosion-resisting washers by a series of helical springs inside

the seal. The effectiveness of the seal therefore does not depend on high pressure; consequently, rubbing friction is low. Because of the bellows construction the seal is self aligning and requires no adjustment for wear.

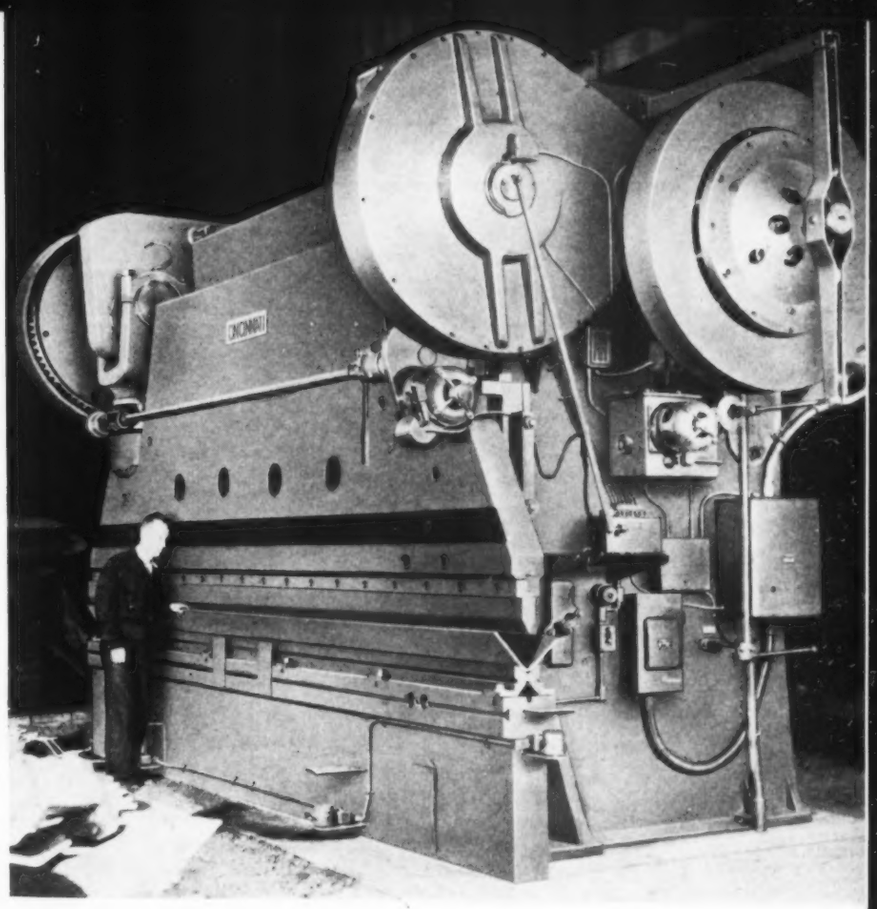
• *Aluminum for Dryers*

Corrosion and rust are accelerated by the hot, humid atmosphere necessarily prevailing around drying machines. In using steel for the dryers, corrosion not only shortens the life of the equipment, but discoloration from rust may also impair the quality of the product. Primarily to eliminate all possibility of corrosion, the Philadelphia Drying Machinery Company has developed a dryer constructed entirely of aluminum.

Helical springs inside the copper bellows seal the cork faces against lubricated sealing rings in Caterpillar tractors

Structural aluminum is used throughout for the frame, with aluminum sheets for all internal partitions as well as the inside faces of all insulating panels. Aluminum bolts are used to assemble the framework. Heating coils and coil supports are also made of aluminum, not only eliminating a source of contamination, but also providing higher heat conductivity and improving heat transfer.

Although this construction is considerably more expensive, the added cost is justified in lower maintenance costs and freedom from corrosion, longer machine life, and the entire elimination of the possibility of contamination.



Electrically-controlled hydraulic piston actuates a friction clutch engaging the drive in the 1,000-ton press brake made by the Cincinnati Shaper Company

• *Press Brake With 1,000-Ton Capacity*

In a further development of the press brake described in *Product Engineering* for March, 1933, engineers of the Cincinnati Shaper Company have developed a design giving safe working loads up to 1,000 tons over a span of more than 18 ft. The welded construction previously described is used in this design with but little change except for the necessary increases in dimensions.

Housings at each end of the press are flame cut from rolled steel 8 in. thick, each piece weighing 33,000 lb. Main drive gears are of welded construction with a high carbon steel rim giving high tooth strength and minimum wear. Welded guards form outboard bearings for the gears. Power is supplied by a 50-hp. high-torque motor driving a 2-ton flywheel through multiple V-belts. Engagement of the drive is controlled by a friction clutch in the flywheel operated by a small hydraulic unit mounted on the side of the housing. Automatic lubrication to all bearings is furnished by two units, one mounted on each housing. Lubricators are driven by a link connected to a small crank on the ends of the press eccentric.

Electrical control stations are connected with cut-out switches so that the machine can be operated from any one station, or may require contact to be made at each station. Controls allow operation with a complete cycle, continuous running or jogging.

To protect the press frame from overload, a direct-reading load indicator is built into the right housing at the center of the throat. This indicator not only shows the load developed during any operation of the press, but a maximum load hand records continually the heaviest load to which the machine has been subjected at any time.

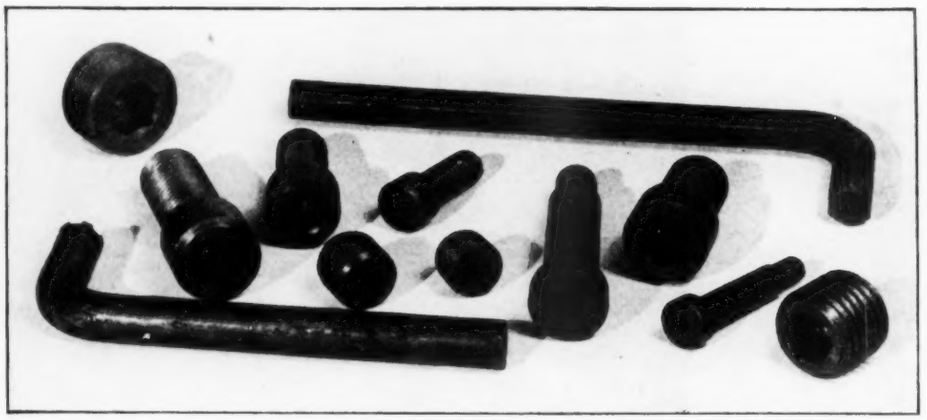
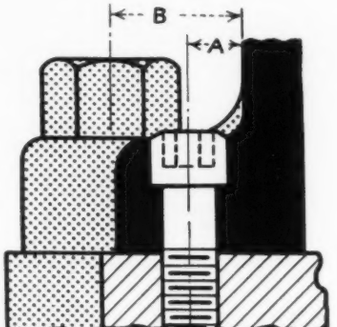


Fig. 1—Types of heat-treated alloy steel hollow set and socket-head capscrews and wrenches

Fig. 2 — Comparison of flange overhang when using carbon steel hex-head and heat-treated alloy steel socket-head capscrews



Advantages of Using Hollow and Socket-Head Screws

Because of numerous advantages over slotted, hexagon and square head screws, hollow set and socket-head capscrews are being used more extensively on applications where holding ability, strength, safety, compactness and appearance are important

MANY of the possible advantages obtainable through the use of hollow-set and socket-head capscrews do not relate to the design of the screw itself, but are reflected in the design of the parts on which the screws are used. By virtue of their greater strength and holding power, these screws can be used without locking devices on internal as well as external mechanisms. Not only can they be used in confined places where it would be inconvenient to use other types of screws because of assembling difficulties, but they make more compact constructions possible. By elimination of projecting screw heads, danger to operators is overcome, flush surfaces are obtained, symmetry and unbroken contours are possible and neatness results. Whereas short screws of small diameters with square, hex or slotted heads are usually difficult to assemble, the wrenches used to drive and set hollow and socket-head screws facilitate speed and assure a tight assembly.

Because hollow and socket-head screws are made of chrome-molybdenum, chrome-nickel or other alloy steels and are heat-treated, they are much stronger than those made of ordinary screw stock. Therefore,

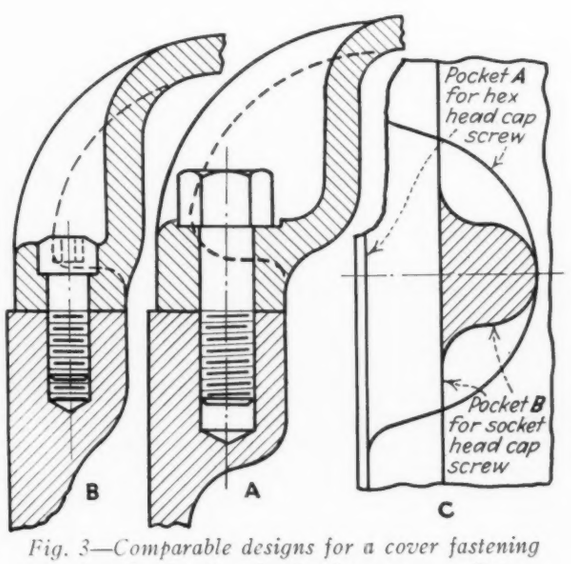


Fig. 3—Comparable designs for a cover fastening using hex-head and socket-head capscrews. Superimposed views show to scale saving of space and weight by virtue of smaller pocket sizes required

smaller sizes can be used than would be possible with untreated carbon steel screws. This is strikingly shown in the accompanying table in which are listed comparative sizes for screws of equal strength, made of heat-treated alloy steel and untreated carbon steel. The figures given apply to both coarse- and fine-threaded screws. It is not unusual for heat-treated alloy steel hollow-set and socket-head screws to withstand a tensile stress of 250,000 lb. per sq.in. This is especially true for setscrews where less ductility is permissible, because stresses are compressive rather than tensile as in capscrews.

One of the immediate advantages of socket-head capscrews is that they permit more compact design. In Fig. 2, a design using a socket-head screw is superimposed over a design using a hex-head screw, both screws being of equal strength as shown in the table. This illustration shows how compactness is accomplished with a corresponding saving of weight. Dimension *A* for the socket-head capscrew is less than 50 per cent of dimension *B* for a hex-head screw. The twisting and bending moment on the flange will be in the same ratio, thereby possibly permitting less flange thickness when using socket-head capscrews.

Another example of greater compactness obtained through the use of socket-head screws is shown in Fig. 3 where pockets *A* and *B* on a cover are required to accommodate the screws. The indentation is smaller and neater when using a socket-head capscrew in preference to a hex-head capscrew. Pocket *A* is necessarily wider and deeper to make room for an "S" or socket wrench. The two constructions are superimposed for an easy comparison of the pocket sizes and the resulting space saving.

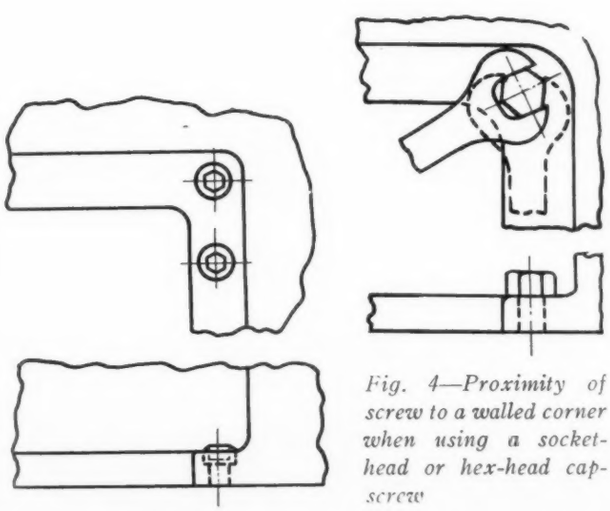


Fig. 4—Proximity of screw to a walled corner when using a socket-head or hex-head capscrew

When a screw is to be placed in a walled corner, as shown in Fig. 4, the deciding factor controlling the proximity of the screw to the corner is the wrench, as shown, or the diameter of the socket when a socket wrench is used for tightening. Because of the smaller screw-head size and the type of wrench used, the head of the socket-head screw needs only to clear the wall, making possible a narrower flange and a tighter joint.

Examples illustrating the advantages of strength are

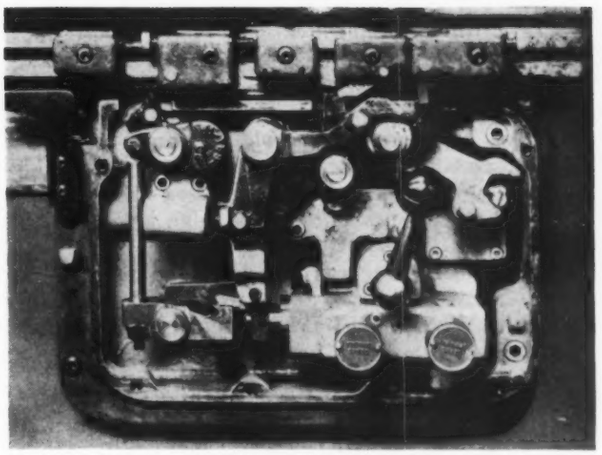


Fig. 5—With the absence of protruding screw heads, more compact design is obtainable. Although no locking devices are used, these screws can be reliably used for fastening internal as well as external parts

Screw Sizes for Equal Tensile Strength

(All sizes given in inches)

Carbon Steel Screws	Alloy Steel Screws	Carbon Steel Screws	Alloy Steel Screws
Not Heat-Treated	*Heat-Treated*	*Not Heat-Treated*	*Heat-Treated*
$\frac{1}{2}$	$\frac{3}{8}$	$\frac{7}{8}$	$\frac{5}{8}$
$\frac{5}{8}$	$\frac{7}{16}$	1	$\frac{3}{4}$
$\frac{3}{4}$	$\frac{1}{2}$		

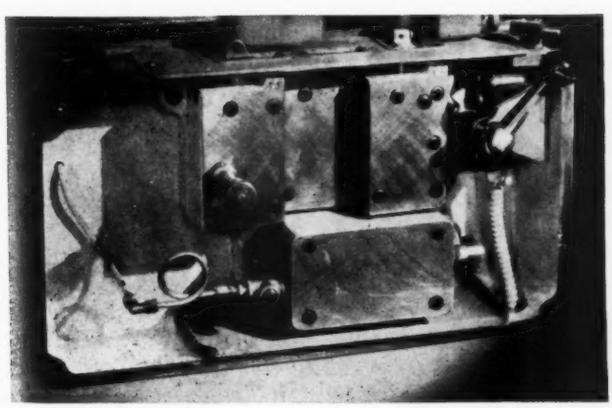


Fig. 6 (Left)—When tight clamping against heavy pressures is required the high strength of socket-head capscrews is of advantage

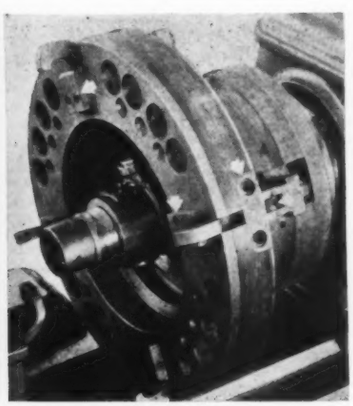


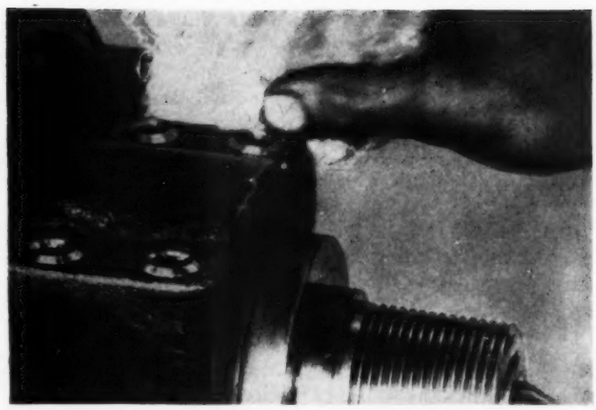
Fig. 7 (Right)—With flush surfaces on rotating parts, the danger from protruding screw heads is eliminated

given in Figs. 5 and 6, in which the table reversing dogs, control box and hydraulic valve cases of an internal grinding machine are fastened with socket-head capscrews. It is especially important that internal mechanisms be securely fastened to eliminate the possibility of breakage. The holding power of socket-head capscrews is sufficient so that a screw locking device is unnecessary. In the valve cases of the hydraulic system shown in Fig. 6, high holding power of the screws is essential because a loose joint would admit air into the line and thereby cause failure of the system. Pipe plugs are also used in the installation in Fig. 6. These plugs are tightened with the same type of wrench and are heat-treated to resist surface wear, an important factor for a sound joint in a hydraulic system.

The elimination of projecting heads is not only a matter of appearance and safety, but the attending saving of space requires screws to be flush with the surface of the part. Thus the use of socket-head capscrews on rotating chucks as shown in Fig. 7 permits utilization of the full swing of the machine and when used on the face of a chuck allows the finishing-tool support to travel closer to the face of the chuck thereby cutting down overhang. And, of course, the possibility of injury by projecting screw heads is eliminated by having surfaces that are smooth and flush.

Comparing the hollow-set or socket-head capscrew with the slotted type, the latter can only be driven with a screwdriver that must be at least slightly thinner than the width of the slot. But usually this fit is too loose with the result that relatively little driving torque can be exerted on the screw. Burrs or slivers are raised on slotted screws when they are being tightened. These are dangerous and must be dressed off after the screw is assembled. Because worn edges are unsightly and are taken as an indication of poor workmanship, slotted screws must necessarily be replaced before the new machine or part is shipped out. But when setting up a hollow or socket-head screw there is no possibility of using other than the proper wrench.

Symmetry and smooth flowing lines are always an asset on any machine, but these may be ruined by projecting screw heads. Even though such a head be sunk below the surface of the part being fastened so as to obtain a flush surface, the counterbored hole must be large enough to accommodate a socket wrench, thereby leaving an unsightly, dirt-catching hole around the screw head. But, by using a socket-head screw, the counterbored hole is filled by the screw head and a smooth, flush surface is obtained. This also has the attendant advantage that when machines are being cleaned, there are no projections to catch the loose threads of the waste as shown in Fig. 8. This handicap is overcome by the use of socket-head capscrews.



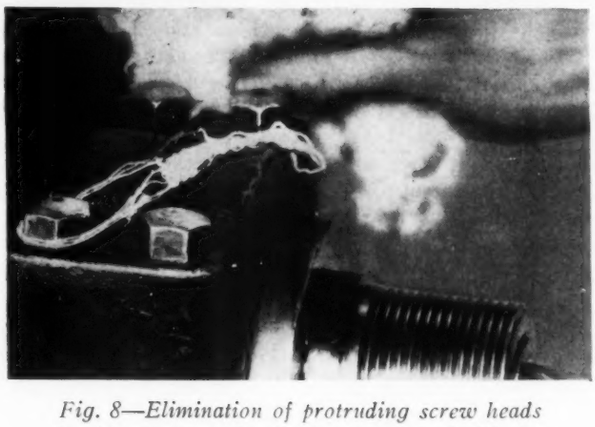


Fig. 8—Elimination of protruding screw heads by the use of socket-head capscrews, facilitates the cleaning of machines

Hollow or socket-head screws are not limited to one or two diameters as on setscrews or fillister-head screws. Head shoulder screws having three diameters, obtainable in numerous body lengths, are used as bearing studs for links, levers, gears, cam attachments and other parts as illustrated by the two methods in Fig. 9. Three diameter socket-head screws are also used in punch and die work as stripper bolts for spring guidance.

Tiny slotted setscrews are difficult to assemble, especially in close quarters. Using a fluted wrench having four driving surfaces, No. 5 screws (0.125 diam.), which are the smallest made, can be placed on the end of the wrench and assembled with ease. The screw cannot slip off the wrench end as a slotted screw would, a highly important advantage especially in these smaller sizes.

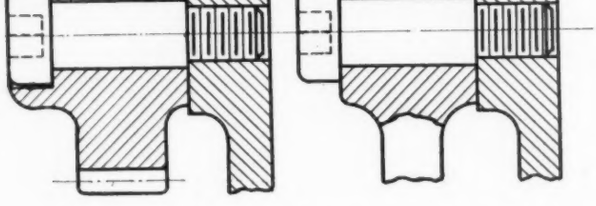


Fig. 9—Standard socket-head shoulder screws may be used in place of special studs for mounting gears, levers, links and other attachments

Cooperation of the following companies, in preparation of this article, is acknowledged: The Allen Manufacturing Company, The Brighton Screw & Manufacturing Company, The Bristol Company, The Holo-Krome Screw Corporation, Mac-it Parts Company, and Standard Pressed Steel Company.

Simplified Velocity and Force Diagrams

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A graphical method for determining the velocities of different points in a mechanism and how to use the diagrams for finding the forces that hold the mechanism in equilibrium

IN THE commonly used method for graphically determining the velocities of various points in a mechanism, it is first necessary to establish the instantaneous center of rotation for each link. If the velocity of one point in the link is known, the velocity of any other point can be found. It will be proportional to the distance from the instantaneous center and in a direction perpendicular to the radius from the point to the center.

It is not always convenient to make the necessary constructions for locating all the instantaneous centers. Often it will require a cumbersomely large drawing. But for links pivoting around a fixed center, the instantaneous center will be the fixed center for all positions of the link. By making use of this fact and the velocity relationships, as stated below, the velocities of all points in the plan of the mechanism can be determined graphically with a minimum amount of time and labor. And by the application of an additional simple relationship, the forces that hold the mechanism in equilibrium can be readily calculated for any position of the mechanism.

The procedure for constructing the velocity diagrams is based upon the following three rules:

1. If a link of a mechanism, such as shown in Fig. 1, rotates around a center O, the direction of the velocity of each point of the link (A, B or C) is normal to its radius (OA, OB or OC) and the magnitude of the velocity is proportional to the length of the radius.

2. The velocity of a slider in a fixed curved guide, as in Fig. 2, is directed along the tangent to the path. In a straight guide, as in Fig. 2, the direction of the velocity is along the guide.

3. For two points on a link, such as A and B in Fig. 3, having velocities v_a and v_b, the projections v_a and v_b of the two velocities on the line AB connecting A and B must be equal. If these velocities were not equal, point A would approach or lag behind point B, which would require that AB contract or stretch. But AB is a rigid member and therefore the distance AB must be constant. In the auxiliary diagram, Fig. 3, the velocities of A and B are represented in a vector diagram drawn from the pole p. The vector heads a and b

FIG.1

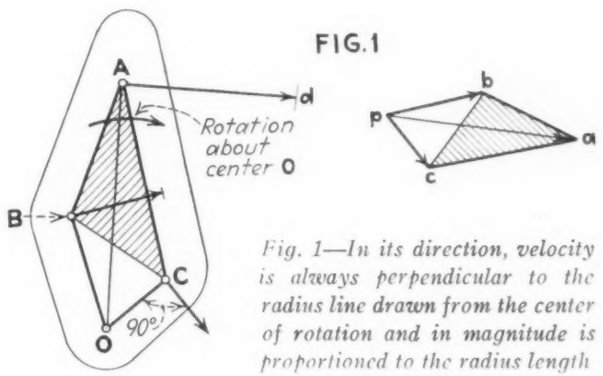


Fig. 1—In its direction, velocity is always perpendicular to the radius line drawn from the center of rotation and in magnitude is proportioned to the radius length

FIG.2

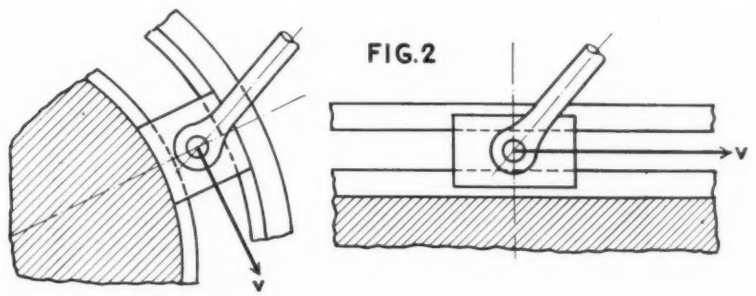


Fig. 2—When traveling in fixed guides, the direction of the velocity will be tangent to the path of travel. For straight guides the velocities will be parallel to the direction of the guiding surfaces

of the velocity vectors *pb* and *pa* are on a line perpendicular to *AB*, as indicated.

Referring to Fig. 3, knowing the velocities of points *A* and *B*, the velocity of any other point on the link, such as *C* or *D*, is found on the diagram as follows:

The projections of velocities v_a and v_b on line *AB* must be equal. The vector heads *a* and *c* must therefore be on a line perpendicular to *AC*; similarly, the vector heads *b* and *c* must be on a line perpendicular to *BC*. The intersection of the two gives point *c*, and vector *pc* is the velocity of point *C*. In the same way, *pd* is the velocity of *D*. Note that the figure *abcd* is similar to *ABCD* on the link, but is turned through 90 deg. The vector heads form a figure similar to the configuration of the corresponding points on the link itself. Therefore, the velocity of point *e*, say at $\frac{1}{3}$ the distance from *B* to *A*, is found by dividing *ba* into three parts and drawing *pe*, which will be the desired velocity vector.

The above considerations are sufficient to draw the velocity diagram for any link mechanism in any position, if the velocity of one point is known. As an example, in the mechanism shown in Fig. 4*a*, the crank O_1A rotates at 120 r.p.m. and is 8 in. long. The velocity of pin *A* is therefore 100.4 in. per sec., as found by calculation. The velocity of point *F* is required. The steps are as follows:

(a) Lay off, as in Fig. 4*b*, the velocity of point *A* to any chosen scale, plotting the vector *pa*, perpendicular to O_1A, as shown.

(b) Find velocity of *B*. Velocity of point *B* must be normal to O_2B, and the vector heads *b* and *a* are both on a line perpendicular to *AB*, as shown.

(c) Find velocity of *C* by drawing lines perpendicular to *AC* and *BC* through *A* and *C*, respectively.

(d) Find velocity of *E*. Velocity of *E* must be parallel to guide *DD*; draw a line parallel to *DD* through *p*. Vector heads *e* and *c* must be on a line perpendicular to *EC*, as indicated.

(e) Find velocity of *F*. Divide *ce* in the proportion of *CF/FE*. Vector *pf* is the velocity of *F*; scaling it off, $v_f = 108$ in. per sec.; in the direction given by *pf* the velocity vector.

Should the mechanism embody slider elements between links, such as in the well-known shaper return element shown in Fig. 5*a*, an additional rule must be considered. In order that the sliding surfaces will not separate or cut into each other, the velocities of the contacting points A_1 and A_2 must have equal components on the line *NN* normal to the contours at the point of contact. If the velocity V_a' of point A_1 of crank OA_1 is known, the velocity v_a'' of the point A_2 on arm O_2A_2 is found, as shown in Figs. 5*a* and 5*b*, by the vector diagrams.

In Fig. 5*a* the vector A_1V_a' representing the velocity of point A_1 on the crank is drawn perpendicular to the crank radius O_1A_1. Similarly the vector A_2V_a'' is drawn perpendicular to A_2O_2, the line joining A_2 with O_2, the center of rotation of the arm. The component of these velocities along the line *NN* must be equal, the vector heads V_a' and V_a'' both lying on a line perpendicular to *NN* which is normal to the slider guide. Thus, by this simple graphical method, if the velocity of one of the points, A_1 or A_2, is known, the velocity of the other point can be obtained readily.

With this information not only can the velocities of any point in a mechanism be found easily, but the velocity diagrams can also be used as a convenient method for analyzing the forces acting on the mechanism. The equilibrium of a mechanism is usually analyzed by the resolution of each force into components and by considering the equilibrium of each link separately. This is sometimes an expeditious method, but usually it is tedious and conducive to errors. An old

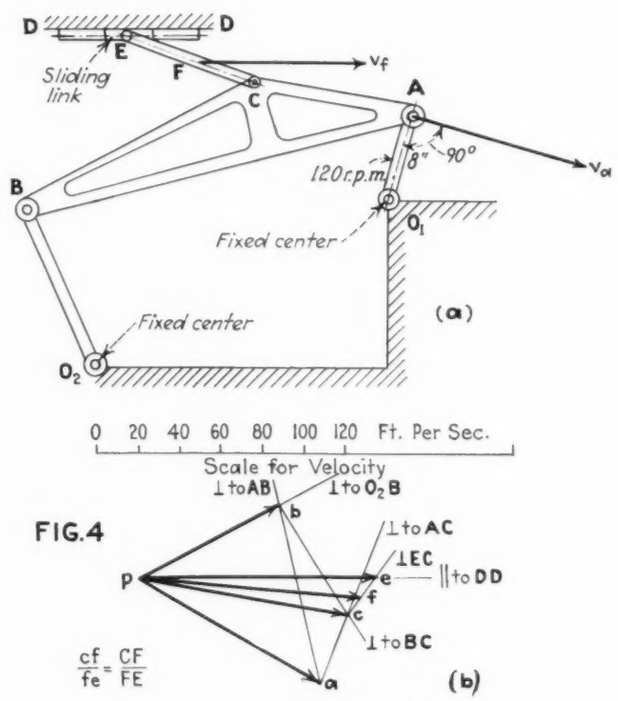


FIG. 4

Fig. 4—Successive steps in the construction of the velocity diagram for a mechanism with five moving links

FIG. 3

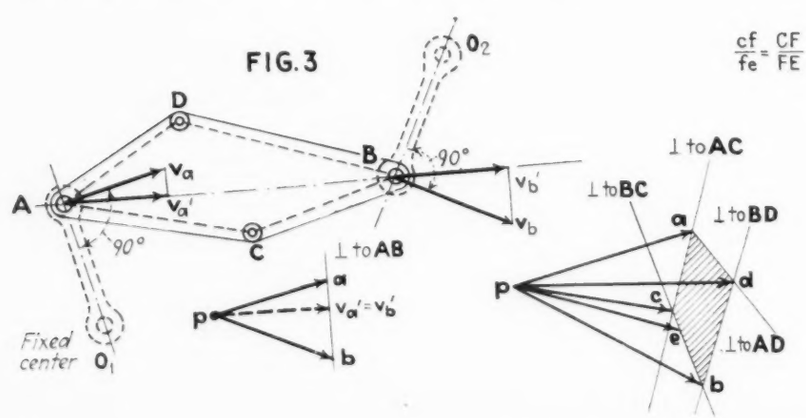


Fig. 3—In any mechanism made up of rigid members, the component of the velocity along the center line of the link must be the same for all points

but not commonly mentioned law of mechanics states that:

"If a mechanism is in equilibrium under the action of a number of forces, and the mechanism is given a small displacement from its equilibrium position, the total work done by all forces is zero."

Referring to Fig. 6, the work of each force is equal to the displacement S of the point of application times the projection F' of the force F on the line of direction of the displacement. The product is positive when the sense of the projection F' and displacement S coincide as for the forces F_1 and F_2, and negative when the the senses are opposite to each other as for force F_3. The mechanism in Fig. 6*a* is in equilibrium if:

$$S_b \times F_2' + S_c \times F_1' - S_d \times F_3' = 0 \qquad (1)$$

The first two terms of this equation are positive, and the third term is negative, in conformance to the rule given.

To translate velocities of displacement into distances displaced, assume that point A moves with an arbitrary velocity through a small interval of time to achieve the small displacement of the mechanism. Then the displacements of all points of application of the forces will be proportional to their respective velocities. Moreover, the directions of these displacements will be the directions of the velocity vectors. Equation (1) can therefore be written in the convenient form:

$$v_b \times F_2' + v_c \times F_1' - v_d \times F_3' = 0 \qquad (2)$$

In this equation v_b, v_c and v_d are the velocities of B, C and D and F_2', F_1' and F_3' are the projections of the forces F_2, F_1 and F_3 on the direction of the respective velocities. The equilibrium of the mechanism is then analyzed as in Fig. 7. The velocity diagram is drawn to an arbitrary scale (v_b is taken equal to 5 units), the projections of the forces on the respective velocity directions are found, and the table is then set up as follows:

Point	Force	Projection	Velocity	Work Products
B	$F_2 = 14$	$F_2' = 9$	$v_b = 5$	45
C	$F_1 = 16$	$F_1' = 15$	$v_c = 3.2$	48
D	$F_3 = X$	$F_3' = X'$	$v_d = 5.3$	$-5.3X'$

Using the above tabulated values as indicated by Equation (2):

$$v_b \times F_2' + v_c \times F_1' - v_d F_3' = 45 + 48 - 5.3X' = 0$$
$$5.3X' = 93$$
$$X' = 17.6 = F_3'$$

This means that if the mechanism is to be in equilibrium, the force F_3 should be such that its component F_3' will be 17.6 lb. By proceeding in the manner as outlined, the velocities and forces acting at any point on a mechanism can be determined.

FIG. 5

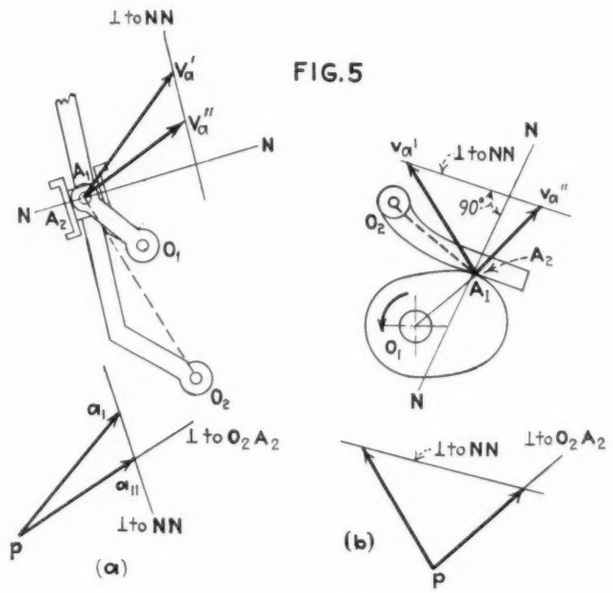


Fig. 5—With the velocity of crank O_1A_1 known, the velocity of point A_2 on link O_2A_2 can be found

FIG. 6

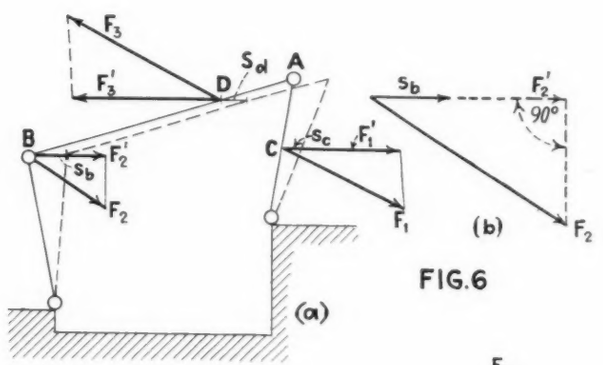
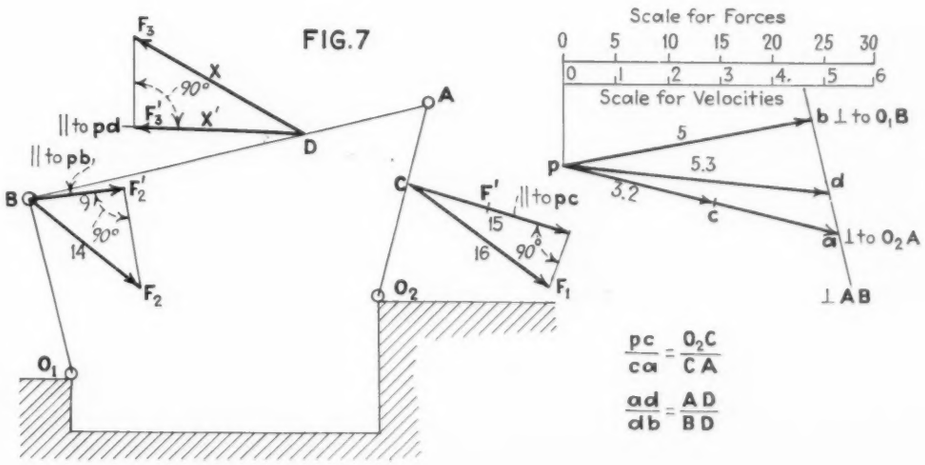


Fig. 6—For a given direction, the work done by each force is equal to the product of the force times the displacement. Work of force F_2 and F_1 is positive; work of force D is negative

FIG. 7



$$\frac{pc}{ca} = \frac{O_2C}{CA}$$

$$\frac{ad}{db} = \frac{AD}{BD}$$

Fig. 7—Example of a velocity diagram for the determination of the equilibrium forces acting on a mechanism

How Solidification Affects Steel Casting

Factors governing the direction of metal solidification, as controlled by the foundry, require definite provisions in the design of the part. Hence, for sound and economical castings, close cooperation between designer and foundry is essential

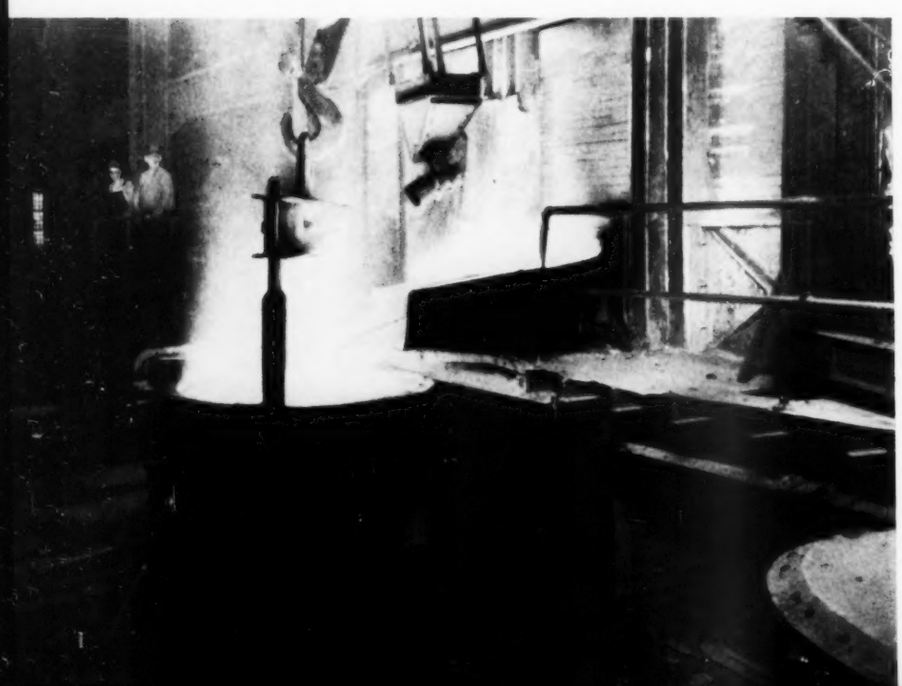
WITH the exception of metallurgical problems associated with the casting of steel, all other steel foundry problems are directly associated with and controlled by design. This controlling factor is not often in the discretion of the founder. The nearest approach to an ideal condition is when the designing engineer consults the foundry during the formation of the design, or when the designer is thoroughly conversant with the behavior of molten metal. Cooperation is not covered adequately by an agreement upon certain minor points affecting the casting, such as increased fillets, satisfactory draft and finish allowances, important as these all may be.

Violent changes or differences of section in the same casting cause trouble in the foundry, but what is most necessary to a designer who desires perfect examples of foundry art, is an understanding of the phenomenon of progression of solidification, its direction and how these phases are controlled. As an ideal example, in a tapered chill-cast steel ingot, cast with the large end uppermost, solidification progresses smoothly and without interruption from the bottom to the very top. There will be a sufficient amount of metal at the top to take care of the liquid shrinkage. The temperature gradient is perfect, the increasing section keeps solidification constantly progressive and in one main direction. Knowing by experience that such ideal conditions can rarely be possible, it is well for the designer to submit his design to the founder for comments and suggestions.

What are some of the principles over which the founder is so concerned, that are often overlooked when designing steel castings? A design should be so sectionally constituted that solidification can be directionally controlled, and thus made to progress toward one or more points located in such a way as to allow for adequate feeder attachment. There is a tendency to exaggerate the difficulties of achieving this, but in most instances it can be accomplished without serious difficulty. External chills often can be used to neutralize any local deviations from the ideal design for progressive solidification.

To illustrate the desired condition, Fig. 1 shows a vertical cross-section of a hub casting of plain type. Such a casting, usually made in green sand, would be gated at *X* in order to avoid sand erosion, and would have attached feeders at *O* to supply the necessary metal for proper soundness. When the length *O-X* is great in relation to the diameter, as frequently occurs, it will be obvious that, with a uniform section, the hottest metal is at *X*, and the metal in the risers, having parted with much of its heat during passage through the mold, will be the coldest. This is a complete reversal of the conditions necessary to produce a sound casting with minimum waste of steel and maximum certainty.

In Fig. 2, with the correct pattern shape, and a slight progressive tapering of the core, a section will be produced which will solidify from the bottom up, and which will require a minimum of feeding to produce soundness. The heat



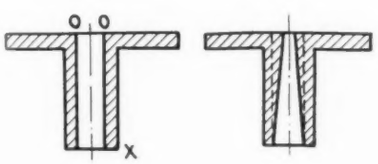
Design

FREDERICK A. MELMOTH

Vice-President
The Detroit Steel Casting Company

localization, caused by the method of gating forced upon the founder by sand conditions, is progressively balanced by the increasing size of the section. Although a simple form has been shown for an example it is evident that in the design of more complicated parts a study of the sections at various points will allow the adaptation of the idea over a wide selection of castings. The following facts must be known to do this in a satisfactory manner: (1) In what position the part will be cast; (2) At what point the necessary gate or gates will be attached. Apart from such rare instances when the designer himself is so thoroughly grounded in foundry practice as to be able to supply the answers, the foundry is the only source of the information, and cooperation in developing the design can save trouble, pattern-making time and defective castings.

It is realized that in spite of the fact that a manufacturer of forged or rolled parts has practically complete control of his original ingot design, and may so arrange matters that perfect solidification of the most favorable type can eventuate, there still exists a difficulty in obtaining perfect ingots. From this it must be obvious that the steel founder, with no power of choice, runs into a great number of solidification problems. Many of them can be solved only by modifying the design of the part to take care of the solidification properly, as in the hub casting described above.



Figs. 1 and 2—The plain hub (left) is greatly improved by tapering (right)

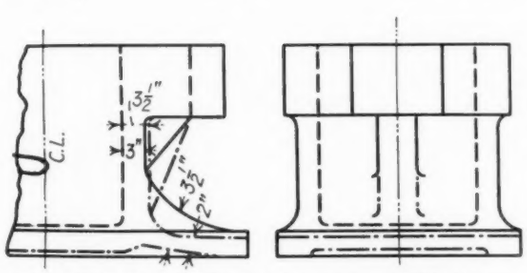


Fig. 3—Cylinder for high pressures. Solid lines are design as submitted to foundry, dot-and-dash lines show changes suggested

In Fig. 3 is shown a high-pressure cylinder, the design of which has been modified to assure progressive solidification. The continuous black lines represent the cylinder as first designed, and the dotted lines show the corrections to produce sections to be fed successfully from the upper part of the casting. It was cast in the position as shown, which was made compulsory by the heavy masses of metal in the upper part. As originally designed the heavy plate section of the base was more massive than the wall and was thereby isolated by the intervening 3-in. wall. Therefore, from the standpoint of solidification, the base section as designed originally behaved as an individual unit.

With the use of chills, there is a liability of shrink cavities forming in the mass at the junction between the plate portion and the head of the cylinder. By removing metal from the plate, and adding an almost equivalent amount to the cylinder walls, and by carrying the heavy buttress down close to the junction of

plate and cylinder walls, a practically perfect metal gradation from the feeding heads throughout the casting was provided. No chills were necessary, and the cylinders made in this manner were perfect in every way.

In Fig. 4 is shown a complicated design of casting commonly asked of the steel foundry, and all the more difficult because of the exacting service for which it is intended. A little thought will show how difficult the insuring of soundness at all parts can be. A further point controlling both the quality of the product and its actual cost is the designing of the part for as simple a molding line as possible, and the elimination of the necessity for loose pieces on the pattern. Accuracy of shape is often as important as soundness.

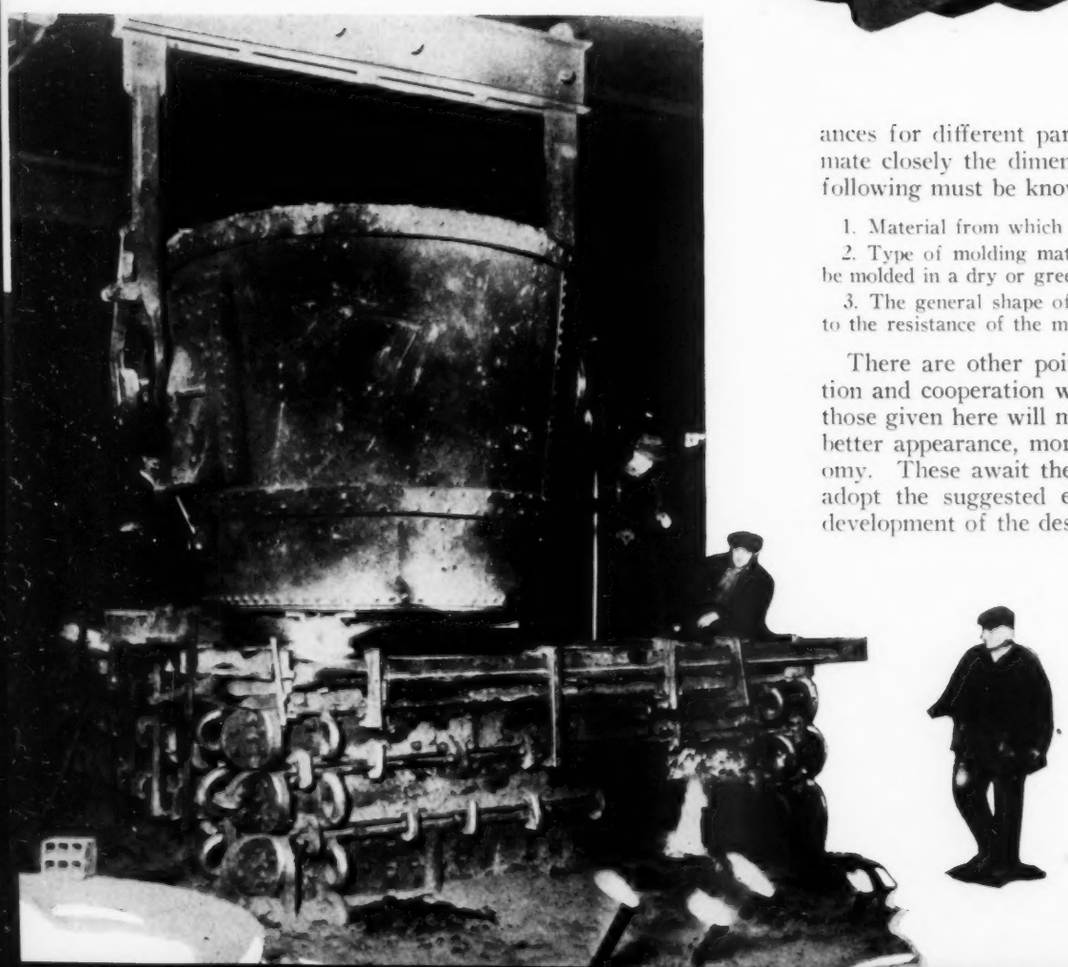
A design calling for a pattern with joints which are difficult to make and maintain, and requiring many loose pieces for pattern parts that cannot be withdrawn directly from the mold, is a fruitful cause of inaccurate castings and, even more so, of dirty castings, owing to the almost inevitable mold disturbances that take place. Many times a design requiring either slow and elaborate hand molding, or difficult coring, could be mounted directly on a molding machine by making insignificant changes of contour.

The final point relative to the necessity for early cooperation between designer and founder is that of insuring the foundry viewpoint upon the question of allowances for contraction. Steel does not have a shrinkage value that is constant for various foundry conditions. Scientific experiments dealing with unhindered contraction arrive at a coefficient of contraction that is a constant. But in the steel foundry, shrinkage is always of the hindered type and the evaluation of the effect of that hindrance can come only as a result of practical experience with a multitude of types of castings, plus an appreciation of the effects of different molding materials and methods of production. It is not uncommon to use two different contraction allowances for different parts of one pattern. To approximate closely the dimensional effects of contraction the following must be known:

Fig. 4—Variation of thickness in wall sections and exacting service requirements complicate founder's problems

1. Material from which the casting is to be made.
2. Type of molding material to be used, and whether it is to be molded in a dry or green condition.
3. The general shape of the casting, and how it will respond to the resistance of the mold during contraction.

There are other points relative to design consideration and cooperation well known to steel founders, but those given here will make possible increased efficiency, better appearance, more satisfaction, and greater economy. These await the user of steel castings who will adopt the suggested early cooperative method in the development of the designs of his castings.



Leather Packings for Hydraulic Systems

GEORGE M. DORR
Dorr-Patterson Engineering Company

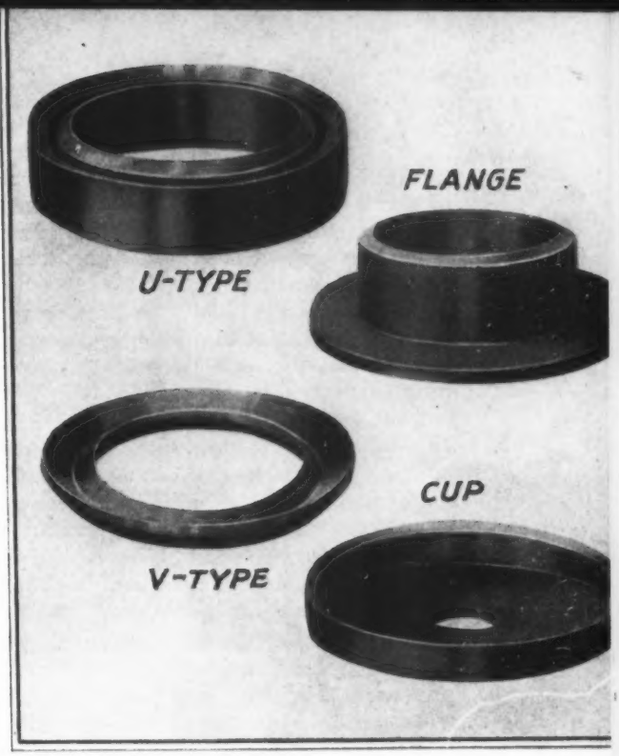


Fig. 1—Four types of leather packings most commonly used in hydraulic systems

Grades of leather and design features for various types of leather packings suitable for different ranges of pressure and conditions of service

NO ONE type of packing material will suit all applications. The most common packing material is twisted or braided flax, hemp or cotton, impregnated with a lubricant and formed into desired shapes. This type is made in long strands that can be cut to the required length. Such packings are low in cost, will withstand moderate pressure, and some will withstand high temperatures; but, they are not suitable for high pressure service, or where excessive friction or wear is objectionable. Some metallic packings have been developed for special applications. Although the metal types are too expensive for general use, they are necessary when open ports or shoulders pass the seal. Metallic packings are also required for service temperatures that are too high for fiber or leather seals.

For normal temperature service, leather is one of the best packing materials because of its excellent wearing qualities, and its ability to withstand high pressure. Because of the action of heat on the gelatine mass in oak-tanned leather, this type becomes hard at temperatures above 150 deg. F., but chrome-tanned and some special types of leather will withstand as high as 200 deg. F. temperature.

Oak-tanned leather is unaffected by liquids ordinarily used in hydraulic systems, but chrome-tanned leather will be decomposed by some types of oils.

Leather in uniform, high quality grades is readily molded to the shapes shown in Fig. 1, but the forming process cannot be done hastily. The heated liquid in which the leather is immersed to attain pliability must completely penetrate the leather. The speed of forming must not be too rapid, and the formed piece must be dried slowly. Finishing cuts can be made only after the leather has been properly treated. If any of these operations are rushed the leather may be burned or the fibers broken, while the packings will open up if the drying or forming is not done correctly.

Although packings cannot be formed to sharp angles without breaking the fibers, leather will assume sharp angles without cracking, when confined under pressure. Packings are formed with a small amount of draft at the rubbing surface to permit a better grip at the seal.

In leather seals, either the hair side or the flesh side can be used against the moving surface. Experience shows that packings used for sealing grease or oil under no pressure should have the hair side against the rubbing surface. The hair side has low tensile strength, but it is close grained and forms a good seal as long as the surface of the leather is not broken. To reduce wear, the metal surface against which the hair side rubs should have a ground finish.

The flesh side is used against the rubbing surface for packings under pressure since this side of the leather is stronger, allows more fiber distortion and withstands the rubbing action better than the hair side. In high-pressure packings effective sealing depends more upon the contact between the beveled lip and the moving part than upon the smoothness of the leather at the rubbing surface. Where sealing is accomplished by forcing the packing against the rod or piston—upsetting the leather by compression or wedge action—the flesh side is placed against the moving part.

The U-type of packing shown in Fig. 1 is efficient throughout a large range of sizes and for both high and low pressures. Since it has a tendency to collapse if not held in shape by pressure, concentric spacers or fillers of metal or leather are placed between the inner and outer faces.

Leather packing of the U-type is supplied in three styles: solid section, open lap, and glued lap. The solid

section is drawn from a ring of leather, and as a result of the forming operation, the inner member is thicker than the outer member, a desirable feature if the inner is the portion that is subjected to rubbing. The range of sizes in which U-leathers can be supplied is limited by the size of the hides. Because of the variations in physical properties of a hide, as shown in Table I, the larger sizes of such packings may not be uniform throughout.

The open lap packing shown in Fig. 2 is formed from a strip of leather with ends scarfed. This type is made from a piece of leather of uniform properties, the section is upset very little in forming, and the fibers are not stressed. As a result, the packing is elastic and uniform in section, and easily applied. Similar properties are obtained in the glued lap packings in which the scarfed ends are glued with a waterproof cement.

The purpose of the filler used in all types of U-packing is to prevent leakage when the joint is not under pressure. The filler also assists in forming the seal when the fluid pressure is applied. Although the filler ring should completely fill the space within the packing, as shown in Fig. 3, the insert should not force the packing against the moving members. The filler should be high enough to protect the beveled edges and so constructed that the fluid under pressure has free access to expand the packing.

Braided hemp, metal or leather are used for filler materials. Hemp expands when exposed to water or oil, and may cause unnecessary friction. Metal fillers are often used. They should fit snugly and be free of sharp edges. If metal fillers are not held firmly the packing will be destroyed under hammer action set up by alternate application and release of pressure. When air or water is the liquid being sealed, the metal fillers should be made from corrosion-resisting materials. For most packing applications leather is the best filler material. Where the thickness must be $\frac{1}{4}$ in. or more the filler is made of two or more layers of leather cemented together with waterproof cement. The cemented joint has the advantage of adding to the rigidity of the filler.

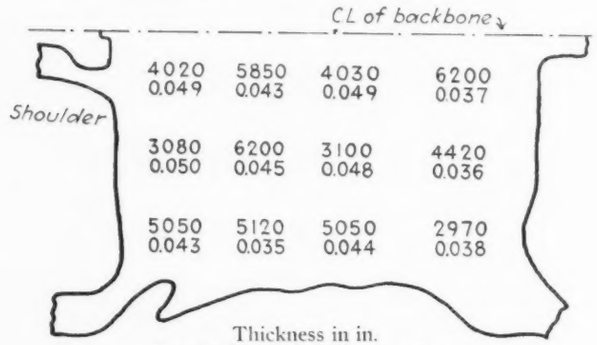
For rods $2\frac{1}{2}$ in. in diameter and smaller the flange or hat packing, shown in Fig. 4, is used in either single or double styles. The single style is suitable for such applications as pump plungers if the pressures are relatively low. Where the service is severe the double style is used, the top lip being forced down upon the beveled lip. This design will withstand pressures in excess of 20,000 lb. per sq.in. when made of heavy leather. Since it makes a more positive seal, this double flange style is widely used for sealing revolving members.

Cup packings shown in Fig. 5 are available for a limited range of sizes. Because of the high cost of application and repairs, this type is seldom used for high pressures if the size is larger than 6 in. in diameter.

Concentric ring packings are used between two members that are bolted or clamped together. This type is also used in gland boxes subjected to high pressures or dirty liquid, or where alignment of parts is poor.

Built-up concentric ring packings, shown in Fig. 6, stand up well under hard service, particularly for revolving members. For severe service and high pressures where it is not important to have a low coefficient

Table I—Variations in Strength and Thickness in an Average Hide



Thickness in in.
Tensile Strength in lb. per sq.in.

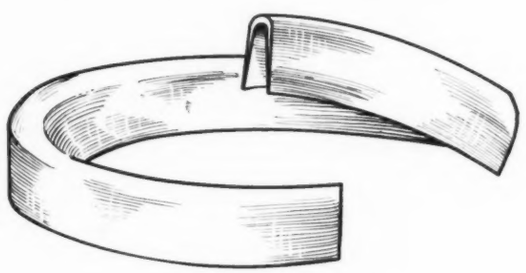


Fig. 2—Large U-type packings may be formed from a single disk or made from a strip as shown here, the joint being lapped or glued

Fig. 4—Flange packings made from leather rings are suitable for small diameters. The double type seal is effective against rotating members

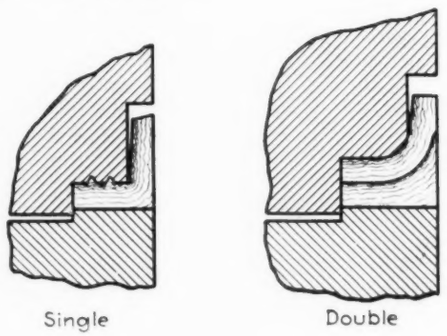
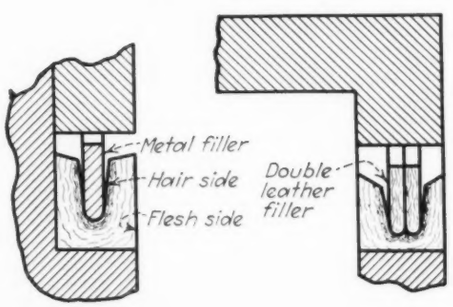
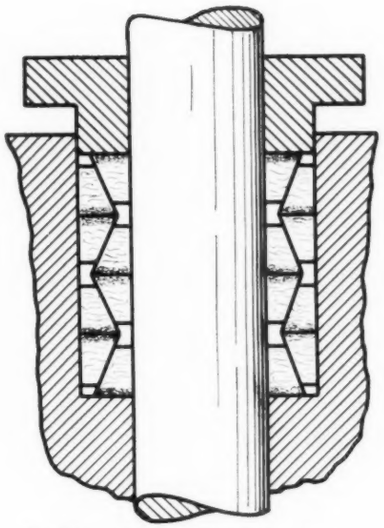


Fig. 3—Filler rings for U-packings hold the sealing edge in place against the moving part





Position of beveled split ring packings before tightening gland

Fig. 6—When compressed the concentric ring packings will seal effectively against high pressure, but friction and wear is relatively high

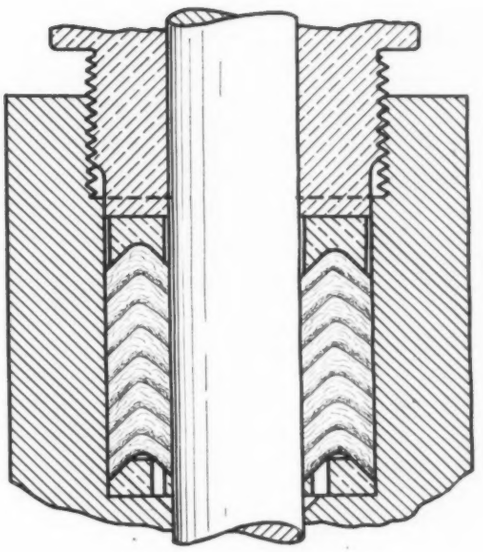


Fig. 7—The multiple V packing has low friction, forms an effective seal against oil or water, can be adjusted to take up wear, and is suitable for heavy pressures

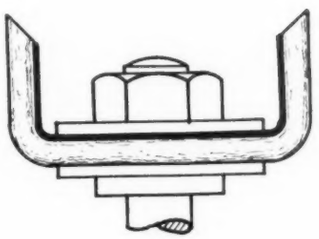


Fig. 5—For small diameters and low pressures, the cup packing formed from a single disk of leather is in general use where replacement cost is low

of friction, concentric ring packings built up of split rings have the advantage of being repaired and replaced easily.

The *V* type packing, shown in Fig. 7, is made up of concentric dished leather rings with formed metal members at the top and bottom. Although this packing is expensive it is highly effective even under high pressure in preventing leakage past the seal. For this reason the built-up type is used on vertical rams where oil leaks are objectionable. Another advantage of this packing is that the glands can be adjusted readily to compensate for wear of the leather. Under high pressures the packing friction is low; when not under pressure the seal is released.

For maximum strength in any of these types of packings, the leather should not be dressed down to a uniform thickness, but the material should be used as it comes from the hide. In leather ordinarily used for packings the thickness varies from $\frac{1}{8}$ to $\frac{1}{4}$ in. The thinnest materials serve well for pressures up to about 150 lb. per sq. in., while higher pressures and more severe service require the maximum thickness.

The added expense of a fine finish on the metal surface that rubs on the leather is a good investment. When the speed of travel is high and the strokes frequent, the application of a polished chromium plate will increase the life of the packing considerably.

Good packings correctly applied should last for several months of continuous service. If a packing fails before this time, the cause of its failure should be ascertained from an examination of the seal when it is removed. Most failures can be traced to one or more of the following causes: Rough surface on the rubbing part, improper installation, foreign matter in the liquid under pressure, misalignment of parts, improperly designed filler or excessive heat.

By examining the leather packing before it is applied, some of the service difficulties arising from the leather itself may be eliminated. The packing should be inspected to see that it is of the proper dimensions, that the hair side of the leather is sound and smooth, and that the bevelled edges are clean cut. In addition, the leather should be of such a tannage and finish that the liquid under pressure will not destroy the packing. When assembled, the filler in a U-type seal should completely fill the packing, and should be high enough to protect the bevelled edge.

Quality Packings

While better prices can be obtained for orders in larger quantities, smaller orders ordinarily will result in less loss during storage when storage facilities are poor. Packings that are well waxed in forming will hold their shape during storage if they are not stacked. To protect the bevel edge each seal should be wrapped in paper and laid flat on a shelf in a dry, cool room. If the packings are not well waxed, they should be placed in wooden molds while in storage to prevent opening up.

In purchasing leather packings, selection of the grade of leather should be made on the basis of performance rather than first cost. In any device that employs leather seals, satisfactory performance on the entire equipment depends upon the quality and design of the packings. Since the cost of the best packing is small compared with the cost of replacing a poor one and the cost of lost machine time, none but the best packings should be used.

COMING

Another article by Mr. Dorr in a coming number will include a check list of essential information for ordering formed leather packings, and correct methods of applying them in hydraulic systems.

Lateral Vibration of Shafts—III

LEWIS F. MOODY
Professor of Hydraulic Engineering, Princeton University
Consulting Engineer, Baldwin-Southwark Corporation

Solutions for two more typical shaft arrangements and approximate methods for calculating the critical speeds are given in this last of a series of three articles appearing in successive numbers of *P. E.*

IN THE previous article it was shown how to calculate the critical speed of shafts with three different typical arrangements of loads and bearings designated as Cases 1, 2 and 3. These and some of the equations developed in the previous articles are referred to in this article. The two additional shaft arrangements, Cases 4 and 5, discussed here, together with those explained previously, cover all the types commonly used.

Case 4—Two Bearings and Three Loads, Two Overhung

This arrangement is illustrated in Fig. 12 and corresponds, for example, to a double-overhung Pelton waterwheel-generator unit, with the waterwheel runners at W_1 and W_2 and generator rotor at W_3.

Referring back to Case 2, for W_3 acting alone, the partial deflections are the same as there developed:

$$y_{33} = \frac{W_3 a^2 b^2}{3\,EIL} \qquad k_{33} = \frac{a^2 b^2}{3\,EIL}$$

$$y_{23} = \frac{W_3 abc_2}{6\,EIL}(L + a) \qquad k_{23} = \frac{abc_2}{6\,EIL}(L + a)$$

$$y_{13} = \frac{W_3 abc_1}{6\,EIL}(L + b) \qquad k_{13} = \frac{abc_1}{6\,EIL}(L + b)$$

With W_2 acting alone, as in Case 2, we get:

$$y_{22} = \frac{W_2 c_2^2}{3\,EI}(L + c_2)$$

$$k_{22} = \frac{c_2^2}{3\,EI}(L + c_2)$$

$$y_{32} = \frac{W_2 abc_2}{6\,EIL}(L + a)$$

$$k_{32} = k_{23} = \frac{abc_2}{6\,EIL}(L + a)$$

Similarly for W_1 acting alone, as in Case 2, we get:

$$y_{11} = \frac{W_1 c_1^2}{3\,EI}(L + c_1)$$

$$k_{11} = \frac{c_1^2}{3\,EI}(L + c_1)$$

Applying the same method for obtaining the value of k_{31} and k_{13}, we get:

$$y_{31} = \frac{W_1 abc_1}{6\,EIL}(L + b)$$

$$k_{31} = k_{13} = \frac{abc_1}{6\,EIL}(L + b)$$

This last equation is the same as that for k_{13}, as given above.

By successive integration the deflection at *1* due to W_2 alone is:

$$y_{12} = \frac{W_1 c_1 c_2 L}{6\,EI} \qquad k_{12} = \frac{c_1 c_2 L}{6\,EI}$$

And similarly:

$$y_{21} = \frac{W_1 c_1 c_2 L}{6\,EI} \qquad k_{21} = k_{12} = \frac{c_1 c_2 L}{6\,EI}$$

The numerical values of the three weights and the six k's may then be inserted in Equation (23), which may be solved for y; and then to get the lowest critical speed the highest of the three possible roots for y is substituted in the equation:

$$\omega_c = \sqrt{\frac{g}{y}}$$

For the case where the dimension and loads are perfectly symmetrical and $a = b = L/2$; $c_1 = c_2 = c$; and $W_1 = W_2$:

$$k_{11} = k_{22} = \frac{c^2(L + c)}{3\,EI} \qquad k_{12} = \frac{c^2 L}{6EI}$$

$$k_{13} = k_{23} = \frac{L^2 c}{16\,EI} \qquad k_{33} = \frac{L^3}{48\,EI}$$

As previously pointed out, the most convenient method of solving Equation (23) is to use trial values, starting with an approximate value of y from an approximate method which will now be explained. A convenient method of finding the final value is to plot the value of the first member of Equation (23) with respect to the trial values of y. Where this line reaches a value equal to the second member of the equation, the desired y is obtained. Two trials are usually sufficient.

Case 5—Three Bearings and Two Loads, One of Which Is Overhung

This case corresponds, for example, to a three-bearing hydraulic turbine-generator unit, with generator rotor W_1 and overhung turbine runner W_2, with the shaft vertical as in Fig. 13.

The partial deflections and k values are found to be:

$$y_{11} = \frac{W_1 a^2}{12\,EILL_1^2}[4L_1^2(L-a)^2 - (L^2 - L_2^2 - a^2)^2]$$

$$k_{11} = \frac{a^2}{12\,EILL_1^2}[4L_1^2(L-a)^2 - (L^2 - L_2^2 - a^2)^2]$$

$$y_{12} = \frac{W_2 acL_2}{12\,EILL_1}(L_1^2 - a^2) \qquad y_{21} = \frac{W_1 acL_2}{12\,EILL_1}(L_1^2 - a^2)$$

$$k_{12} = k_{21} = \frac{acL_2}{12\,EILL_1}(L_1^2 - a^2)$$

$$y_{22} = \frac{W_2 c^2}{12\,EIL}[4L(L_2 + c) - L_2^2]$$

$$k_{22} = \frac{c^2}{12\,EIL}[4L(L_2 + c) - L_2^2]$$

The numerical values of the k's or y's may be found and inserted in Equation (16) or (17), respectively.

Approximate Methods

The first of these methods was proposed by Dunkerley in 1894, and was found by him to agree well with experimental results. It consists in finding the critical speeds N_{c1}, N_{c2}, N_{c3}, etc., due to each load alone. The lowest critical speed N_c of the system is then given by:

$$\frac{1}{N_c^2} = \frac{1}{N_{c1}^2} + \frac{1}{N_{c2}^2} + \frac{1}{N_{c3}^2} + \cdots\cdots$$

This equation reduces to:

$$N_c = \frac{187.6}{\sqrt{y_{11} + y_{22} + y_{33} + \cdots\cdots}}$$

In this last equation the denominator is the square root of the sum of the static deflections of each load acting alone. Dunkerley's method furnishes a short means of obtaining a rough check, but may be considerably in error for complex arrangements, such as a shaft with three bearings and two loads, one of which is overhung, or a shaft with a greater number of bearings.

Remarks on Practical Applications

A high degree of accuracy is fortunately rarely needed in critical speed problems, since it is usually possible and always desirable to keep the highest operating speed well below the critical value. The following are some of the conditions that will affect the accuracy of the result.

Some of the factors which tend to lower the critical speed and to reduce the margin of safety are:

1. Flexibility of the bearing supports. If capable of calculation, the effects of bearing deflections should be added to the deflections of the shaft itself.
2. Weight of shaft itself. This should not be ignored. If fairly small compared to the concentrated weights carried by the shaft, these weights may be increased to include the adjacent portion of the shaft.

Some of the factors which tend to increase the critical speed are:

1. Gyroscopic effect of disks, which change their planes when the shaft deflects. If great, this effect may be computed. See Timoshenko's "Vibration Problems."
2. Attachment of loads by long press or shrink fits. Stodola found that for such fits the effective shaft diameter may be taken as that of the attached hub. Many other questions, such as the effective location of the bearing reactions may sometimes cause uncertainty.

It should be remembered that considerable intermittent vibration may occur before the speed reaches the full critical speed, the margin necessary to avoid such vibration being dependent upon the degree of unbalance of the revolving weights. Sometimes a shaft if accurately balanced may be run at the critical speed without failure or excessive vibration, since the speed does not usually continue long at exactly the critical value. But, in general, the region of the critical speed should be carefully avoided as it involves the danger of serious vibrations or actual rupture.

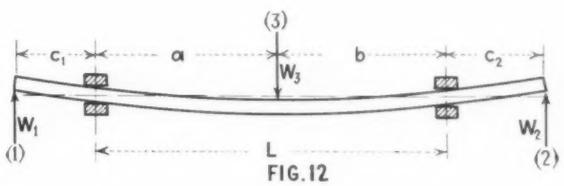


FIG.12

Fig. 12—Two-bearing shaft with three loads, an arrangement corresponding to a double-overhung Pelton waterwheel-generator unit. Fig. 13—Three-bearing shaft with two loads, one of which is overhung, an arrangement found in vertical hydraulic turbine-generator units

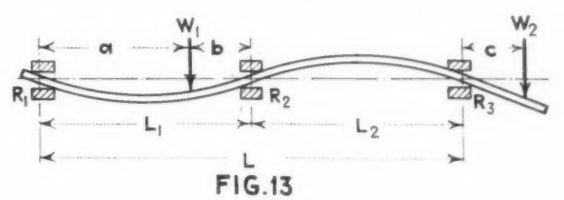
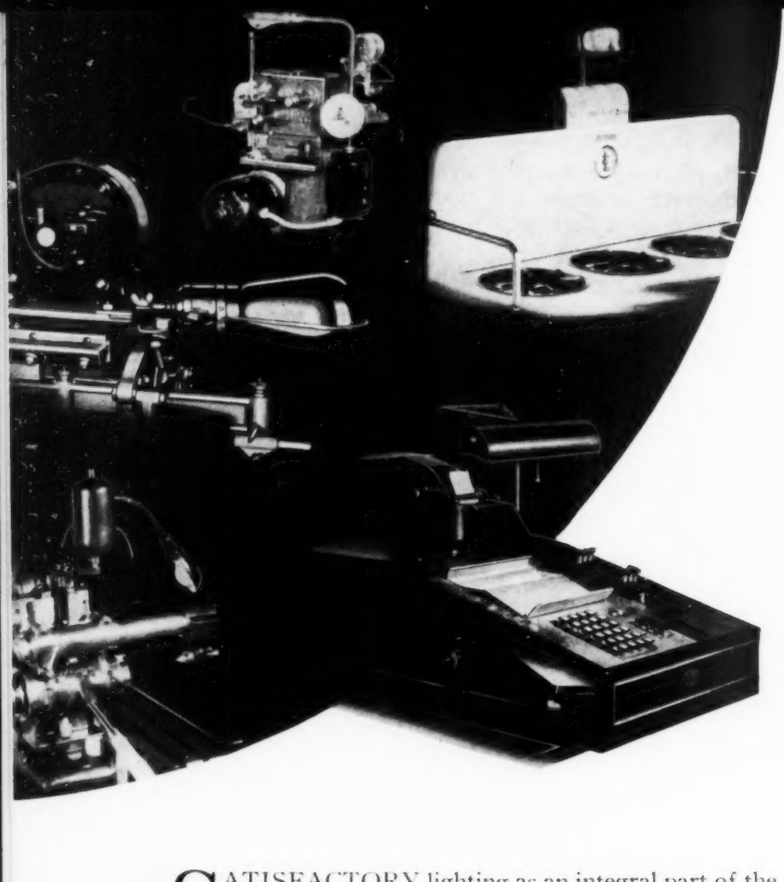


FIG.13



Lamps and

Illumination characteristics for available combinations of lamps and reflectors designed for built-in equipment on all types of machines and appliances

SATISFACTORY lighting as an integral part of the product involves five elements: the light source, the reflector, the wiring device or lamp support, electrical connectors and switches. For machines, appliances and instruments each unit must be selected to meet the special conditions involved in each application.

Factors that must be considered in the selection of a lamp are the intensity required, the distance from the lamp to the work, the presence of vibration and the type of the reflector used. Since reflectors also influence the light intensity, the lamp and reflector combination must be considered together.

Lamps are available with any of four standard bases, but most machine lighting requirements are met by the two smaller sizes—the medium and the candelabra bases. The medium base, which is the standard size for household fixtures, is used for lamps of 10 to 150 watts, while ordinarily the candelabra base is used for smaller bulbs. Where the lamp is placed close to the lighted surface the smaller base with its smaller socket saves space.

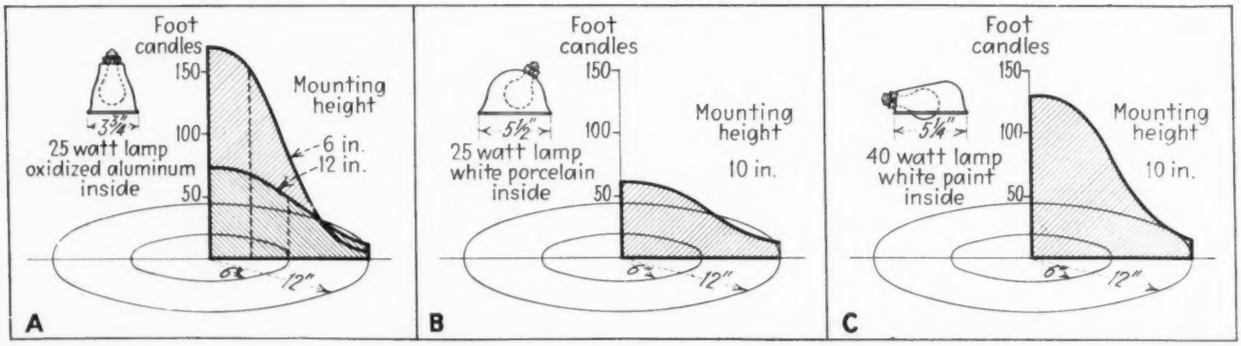
In the selection of the lamp bulb, vibration conditions may require the use of a special type of lamp. Filament life in the standard lamp is shortened by high frequency vibrations such as caused by motor hum or high-speed gearing. For conditions of severe vibration a special type of lamp is available in the 50-watt size having more filament supports and a shorter, stiffer filament. The light output of this lamp is approximately the same as the output of a standard 40-watt bulb. Although the added vibration resistance is obtained at some sacrifice in light output, the life of this special lamp is several times as long as the life of a standard bulb under severe service.

Light output in standard lamps is nearly in proportion to the watt rating, as shown by the following table. The maximum light intensity for each of the six sizes of inside frosted lamps given is for the same type of reflector.

Light Intensity from Standard Lamps

Lamp Rating in Watts	Light Intensity in Foot-Candles		Lamp Rating in Watts	Light Intensity in Foot-Candles	
	Distance			Distance	
	5 in.	10 in.		5 in.	10 in.
15	110	30	*50	340	110
25	220	60	50	430	170
40	350	110	60	590	220

*Vibration resisting.



Reflectors

For Built-In Lighting

With a given combination of lamp and reflector the efficiency of the lighting unit depends on the reflectivity of the inside surface and the shape of the reflector bowl. For the inside surface, white paint has a high reflectivity of 80 per cent and diffuses the light, but does not stand up well where dust and dirt may accumulate. Polished surfaces of aluminum and chromium reflect 65 to 70 per cent of the light, and are well adapted for reflectors that focus or direct the beam. However, polished surfaces deteriorate rapidly in the presence of dirt, smoke or fumes unless the reflecting surface is protected by a glass cover. If a cover is used, light absorption in the glass tends to reduce the efficiency.

A reflecting surface of oxidized aluminum gives good diffusion with a high reflectivity of 82 per cent, but this surface must also be protected from dirt and smoke. Porcelain enamel reflecting surfaces do not collect the dirt as readily as do other surfaces, and because the dirt is easily removed without damaging the finish, the enamel surface is excellent for industrial applications. The initial reflectivity of 70 to 75 per cent is not as high as that of white paint or oxidized aluminum, but the original reflectivity is maintained indefinitely and no protecting glass cover is needed.

To meet differing requirements for light distribution, reflectors are available in various shapes. If a small area is to be illuminated, a concentrating reflector with a small bulb is mounted close to the work. When a large area of high intensity light is needed a reflector with a wide distribution is selected. In order to distribute light uniformly over a larger surface the reflector will be mounted at a greater distance from the work and will also use a lamp of greater power. Thus, the distribution of light over a given working area depends on the shape of the reflector and the distance between reflector and the work.

A reflector with a small diameter and a 25-watt lamp, as shown at *A* in the diagram on the preceding page, throws an intense spot of light over a relatively small area. When the reflector opening is 6 in. above the work the light intensity within a radius of 3 in. is approximately uniform, at a level of about 150 foot-candles. If the reflector is moved away to a point 9 in. from the working area, the maximum intensity decreases, but now the light distribution is approximately uniform over an area within a radius of 6 in. Beyond a radius of 9 in. the intensity at any height of reflector does not exceed 30 foot-candles, an intensity too low for accurate seeing.

To provide a satisfactory level of intensity at a radius of 9 in. or more, a different reflector shape is needed

Added sales value through built-in lighting and how to obtain good illumination, presented in

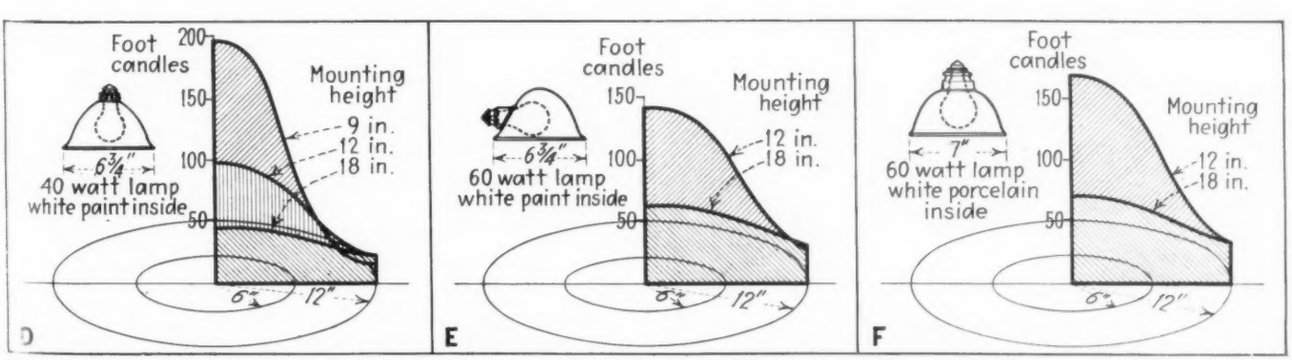
FOUR ARTICLES

1. Good lighting designed into the equipment increases performance and utility—January, "Design for Better Seeing."

2. Minimum requirements for intensity, quality and distribution of light—February, "Lighting Requirements."

3. This article—Illumination characteristics of lamps and reflectors for built-in service.

4. Coming—Data on types of wiring devices, adjustable supports, cords and switches suitable for machine lighting.

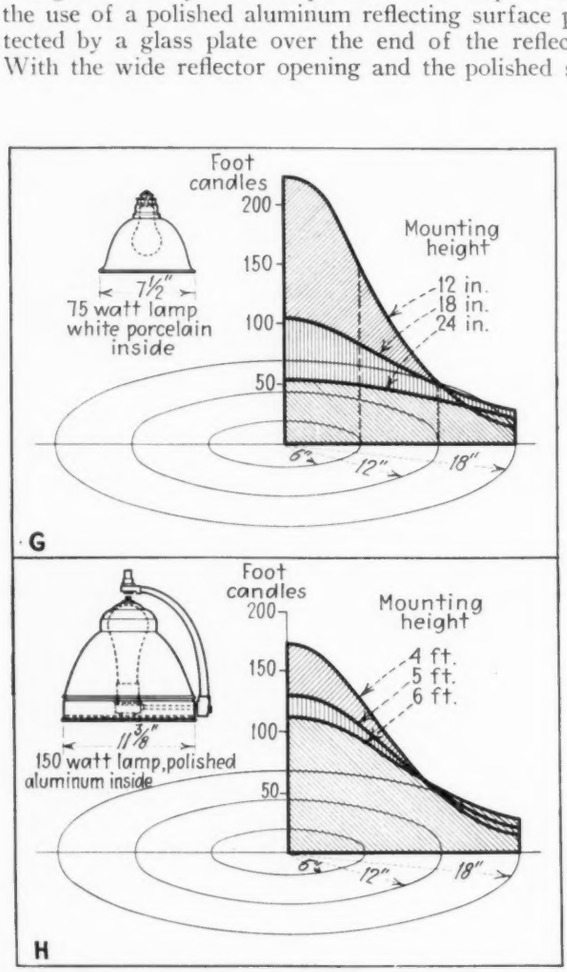


in combination with a lamp of higher power. The combination shown at *F* gives a minimum intensity of 50 foot-candles over a radius of 9 in. at several mounting heights. A high intensity of 150 foot-candles can be obtained over a radius of 6 in. with the reflector opening 12 in. above the work.

Reflectors of the types shown in diagrams *A* to *G* have been extensively used for local lighting on work benches, inspection tables, and other fixed locations. For built-in lighting on machines and appliances, these types of reflectors are well suited to applications where the area to be illuminated is small. But, where larger lighted areas or higher light intensities are needed, lamps of higher wattage must be used. To obtain uniform and efficient distribution of light without glare, lamps of such high light output require the use of special equipment designed for machine applications.

In special lighting devices, such as the unit shown at *H* below, uniform distribution and high intensity are obtained by using a 150-watt lamp and a concentrating reflector. This unit can be placed 5 or 6 ft. away from the work, and still supply a minimum intensity of 50 foot-candles over a radius of 12 in. At a height of 4 ft., a light intensity of more than 125 foot-candles is directed over a circle of 6 in. radius.

Light efficiency in this special unit is improved by the use of a polished aluminum reflecting surface protected by a glass plate over the end of the reflector. With the wide reflector opening and the polished surface, light is converged on the work from a relatively large diameter source, reducing shadows as well as improving distribution.



Standard reflector and lamp (above) and a special lighting unit (below) giving high intensity of light over a large area

Glare from the polished reflector surface is eliminated by a black, non-reflecting louver or ring assembly in the reflector opening, as shown in the accompanying illustration. These louvers are placed so that direct light from either the reflector or the lamp bulb cannot reach the eyes of the machine operators. If necessary the light beam can be directed at the work from in front of the operator without excessive glare.

Each of the various types of reflectors shown in these diagrams is designed for use with but one size of lamp bulb. The bowl is shaped in each reflector with reference to the position of the filament and the dimensions of the bulb. In each design the reflector completely incloses the lamp, allowing no direct light from the bulb to reach the eyes of the machine operator. If the bulb specified is replaced by a larger bulb, some of the light will be scattered, the projecting end of the bulb may cause glare, and the distribution of light is altered. With a smaller bulb than the size specified, the reflector will take up more space than necessary, and, of course, a much lower intensity of light will be obtained.

In the Miller "Suplite," concentric louvers over the reflector opening prevent glare from the polished reflecting surface

The selection of a light unit for a given application therefore involves the selection of the proper combination of lamp and reflector giving the needed intensity, distribution and diffusion of light for best vision.

Once built into the machine or appliance, adequate lighting is assured under any condition of room lighting. Even if the user knows nothing of the advantages of adequate illumination, the precision, capacity and quality of output designed into the machine will be protected. And, a properly designed lamp and reflector is the final step in making the machine a complete, self-contained unit.

Data on the light output of lamps obtained from the Westinghouse Lamp Company. Charts and diagrams for reflectors supplied by the Miller Company.

Equipment shown on the first page made by: Foley Manufacturing Company, Tinnerman Stove and Range Company, International Business Machines Corporation, W. H. Nichols, Alfred Hofmann and Company.

QUESTION AND COMMENT

Anti-Friction Bearings in 1935 Automobiles

To the Editor:

● In the article in February and March describing recent developments in automobile design, many important changes in bearings were not mentioned.

Several important changes have taken place recently in the transmission bearings. All types of bearings—tapered roller, straight roller, ball and needle—are finding wider application in today's transmissions. Plain bearings for the countershafts are being replaced by needle bearings. Reverse idlers are being equipped with straight roller or needle bearings. Tapered roller bearings are being designed into sliding gears. In many cars today, there are no plain bearings in the transmission. Because of the higher load capacity at high speed, the use of anti-friction bearings, together with a change to high-strength alloy steel gear material, has made it possible to retain the transmission designs of two or three years ago for engines of considerably larger power.

Pre-lubricated and sealed needle bearings have become standard for universal joints, replacing all other types of bearings. Requiring no more space than a plain bearing, the needle bearing has the important advantage of requiring no servicing during the life of the car. Therefore, the universal joints can be inclosed and sealed, giving much better protection against dirt and water.

Needle bearings have found several other applications in the new cars because of freedom from servicing. Fan and water pump bearings, steering arm bearings, king pins and clutch linkages are new applications for which needle bearings are being adopted.

Small roller bearings and small ball bearings, sealed for life, are extensively used in some of the steering mechanisms. One maker announces that pre-lubricated sealed anti-friction bearings are used throughout the entire steering system this year. Although a small amount of friction apparently is desirable for stability in most steering systems, the greater steering gear load makes it necessary to reduce friction to a minimum. Also, the sealed bearings eliminate wear at points where safety is of primary importance and eliminate the necessity for lubrication. In the new center-point steering, in nearly all designs, a two-row sealed ball bearing is used to support the center arm.

—R. J. PALMER
New York, N. Y.

Wood Handles

To the Editor:

● In an article by E. R. Harris in the February *Product Engineering*, page 69, mention is made of trouble caused by wooden handles splitting in assembly. Also the molded composition handle is said to be better in appearance than a wooden handle. While it is true that in many instances molded plastic is better both as to appearance and convenience, there are other applications where wood handles serve the purpose best.

Types of wood differ in tendency to split. Some are straight grained and split easily; others have an interlocked grain and split only with difficulty. Such woods as gum, willow, beech, and elm have little tendency to split.

The appearance of small wood handles is dependent upon the method with which the finish is applied. The proper method of finishing such handles in quantity is to use a machine that dips a number of the handles into a tank containing the finish. The machine withdraws the handles at a slow rate, equal to the rate of flow of the finishing material, resulting in a smooth surface of fine appearance.

—JOHN E. HYLER
Peoria, Ill.

Line Contact in Worm Gears

To the Editor:

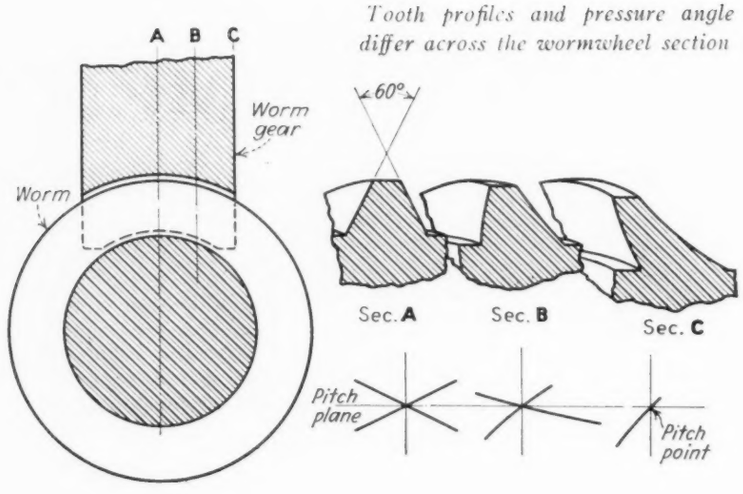
● From his description in the February number, page 67, G. W. Hart is perhaps generalizing about a limiting case. The methods of determining contact location in worm gears were given by Earle Buckingham in *Product Engineering* in a series of articles that appeared during 1930, and by W. M. Pohl in the 1932 February number. These articles assume that the hobs used have their cutting edges in surfaces corresponding to those of the worm, and that cutting and mounting dimensions of the gears correspond.

These suppositions may only be realized in practice when the gears are supported very rigidly. Slight elastic deformations of the gear case and companion parts do occur, changing the theoretical lengths of the contact lines to some extent, but rarely to the extent that spot contact occurs. The contact is still line contact, broadening to band contact under the influence of the elastic deformation of the tooth flank surfaces under load.

However, when the gears are not cut with hobs whose cutting edges lie in surfaces corresponding to the worm surfaces, conditions do change from the above. In a gear cut with a hob with one extra start, initial contact will be a point, increasing with load to spot contact over a wide range of initial adjustment of the gear with respect to the worm.

Before generalizing on this subject it is well to consider the range of variations that may occur in cutting. Contact conditions with hobs for accurate duplication of worm gear tooth shapes are given in *American Machinist* by Nikola Trbojevitch in August 27, 1931, and by R. V. Hutchinson in July 4 and 18, 1934. With precision manufacture, maximum contact can be fixed at the start without having to wait for the gears to wear in to a seat.

The articles by Buckingham and Pohl show what the contact is, while



Tooth profiles and pressure angle differ across the wormwheel section

Trbojevitch and Hutchinson demonstrate ways of attaining the desired contact.

Until cutting and mounting conditions are strictly specified, there is little use in a discussion of this subject, for the contact may be anything from a point, spot or line to a band.

—ROLAND V. HUTCHINSON
Lansing, Mich.

To the Editor:

● The combination of a worm and hobbed wormwheel with a concave face results in a line contact rather than the point contact mentioned by G. W. Hart on page 67 of the February number. The shape and length of the contact line varies from start to end of the contact and is affected also by the nominal pressure angle, helix angle, the position of the pitch plane in the worm depth and the relation between the worm diameter and the circular pitch. Since the number of combinations of these variables is almost infinite the corresponding forms of contact lines and lines of action are also of infinite number.

To facilitate comparison of the contact of the worm and wheel with that of the spiral or helical gear, the helical gear is defined as one formed from a spur gear by twisting the teeth uniformly throughout the face width. When two helical gears are meshed on parallel axes there is line contact, but when the axes are not parallel there is only point contact.

A multiple thread worm, with a helix angle of about 45 deg. may closely resemble a similar helical gear. Structurally and mathematically, the two types of gears may be identical, but when either gear is meshed with a similar gear the contact is very different than if meshed with a concave hobbed wormwheel.

The point of distinction between helical and worm drives is not related to the helix angle of the driver but to the method of producing the mating or driven member of the pair. Two gears in a helical drive are produced by the same tool or method; both have spur gear tooth profiles in planes transverse to the axis. In the hobbed worm and gear pair, however, the driven member is formed by a hob which is identical to the worm. There is therefore no condition such as that mentioned by Mr. Hart as a transition from one to the other. There is no point where with increasing lead and number of threads or teeth on the driver, the combination changes from a worm to a spiral gear drive.

In practice a wormwheel is formed by rotating the hob and feeding the worm blank in a direction either perpendicular or parallel to the worm axis. A study of the contact conditions is facilitated by considering the worm to be a rack which rotates the gear by sliding the worm on the worm axis.

The simplest form of worm and wheel is one having a spur rack profile in the axial plane. In such a worm, if sections are cut which are parallel to the axial section perpendicular to the gear axis as shown in the accompanying diagram, the simple rack profile of the center section is considerably distorted in the side sections, especially in the section most remote from the center. Nevertheless, these rack profiles in the outer sections, unless abnormally distorted, will generate satisfactory conjugate profiles on the infinitely narrow parts of the worm wheel contained in these sections. A lead relatively long in proportion to the worm diameter, giving a large helix angle, results in relatively greater distortion in the outer sections.

Although these rack profiles are not used in design for obvious reasons, mathematically the rack form results in tooth contact as good as that obtained from the simple involute form in the center section.

Lines of action, shown below the sections, become curved due to the curvature of the rack forms, while the pressure angle on one side is greater and the other side less than the normal pressure angle. In section *C* the pressure angle on the low side becomes less than zero and contact is impossible, whereas on the high side it rises to more than 40 deg. On the left-hand side of section *C* there can be no contact, undercutting existing here in the same manner as in spur rack and gear contact.

If undercutting is to be avoided on the side of the thread having the small pressure angle, a high pressure angle is necessary on the opposite side. Any attempt to use a $14\frac{1}{2}$ deg. pressure angle on a worm with a helix angle of greater than about 20 deg. will result in undercut on the low angle side. If the nominal pressure angle is large relative to the helix angle, contact exists simultaneously in all the sections and on both sides of the thread, assuming no backlash.

But this consideration obviously applies only to that type of wormwheel which is generated from a hob similar to the worm. The spiral milled form of wormwheel, although widely used for certain purposes, is only a spiral gear with a point contact, both members of the pair being produced with the same milling cutter.

—ARTHUR L. WILDER
Wethersfield, Conn.

Index Lines on Nomograms

To the Editor:

● On pages 6 and 7 of "Design Work Sheets New Series," A. D. McKenzie has contributed a nomogram of eight scales for computing volume, weight and cost, but his instructions for its use could be made clearer. For instance, his instructions do not show how to obtain the area of a hex bar $\frac{7}{8}$ in. across flats. This information is on the chart, and could be found by adding "on line *C*" after "cost per hundred pieces" at end of first paragraph, and inserting the sentence, "Read area on line *G*," in the sixth line from bottom of second paragraph.

On a chart of this type index lines assist in using the data. In this chart at least six index lines are needed, as shown in the accompanying diagram. These lines have small circles at each significant intersection, the line being discontinued where it passes other scales that are not intersections

KEY TO INDEX LINES

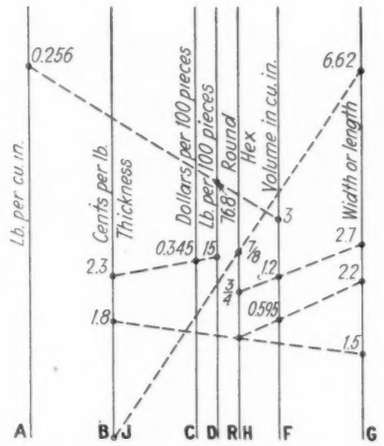
ADF For pieces having a volume of 3 cu.in. of material weighing 0.256 lb. per cu.in., the weight per 100 pieces is 76.8 lb.

BCD 100 pieces weighing 15 lb. per 100, at 2.3 cents per lb. will cost 34.5 cents.

BHG A rectangular strip 1.5 in. x
HFG 0.18 in. and 2.2 in. long contains 0.595 cu.in.

RFG A $\frac{3}{4}$ in. round bar 2.7 in. long contains $\frac{1}{2}$ cu.in.

JHG Using 0.1 on *J*, the area of a hex $\frac{7}{8}$ in. across flats is 0.662 sq.in.



To assist in using the chart, index lines are drawn for typical examples. Intersections used in the examples above are circled

Index lines and examples for assistance in using this chart are given in the accompanying table. To eliminate confusion *J* is used for the right side of scale *B*, and *R* and *H* are the right and left side of scale *E*. The "per hundred" figures may be more convenient for shop practice, but they are somewhat confusing.

—CARL P. NACHOD
Louisville, Ky.

Bearing Lubrication

To the Editor:

● In the article "Lubrication of Anti-Friction Bearings" by F. Leister in the January number, the effect of windage on oil is mentioned. In large exhaust fan bearings, installed in woodworking plants for collecting and exhausting shavings, sawdust and chips, the effect of windage is just the opposite of the effect stated in the article. Unless the housing is sealed, there is a direct suction along the shaft which draws oil from the bearings. The remedy ordinarily used is a deflector mounted on the shaft adjacent to the bearing. In this application, grease is undoubtedly a much better lubricant, as it can not be drawn along the shaft so readily.

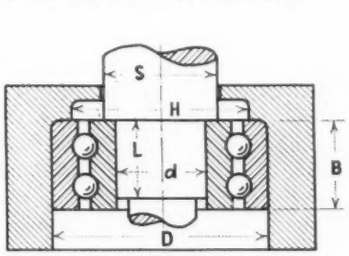
Lack of mobility in grease lubricant would seem to make it especially safe and desirable for use in the lubrication of electric motors to prevent lubricant from finding its way into the motor windings. However, there are too many maintenance men who operate a grease gun on the theory that where a little is good, more is far better. A man with the grease gun fever is very likely to force grease out of the bearings, and the excess grease will be pushed along the shaft and on the windings. Once deposited on the windings, it is perhaps worse than oil.

—J. E. HYATT
Rockford, Ill.

Tabulating Ball Bearing Dimensions

DAVID FLIEGELMAN
Coppus Engineering Company

● For most ball bearing applications, designing time can be saved by the use of a table of shaft and counterbore dimensions for the bearings most frequently used. In addition to its convenience, such a table also insures consistency on all drawings. This table can take the form reproduced here, listing machining dimensions of shafts and housings for each inside and outside diameter of a group of bearings. The dimensions in this table are selected to give a press fit on a rotating inner race with the outer race free to creep. For other types of fits the form of the table will be the same.



d Bearing	Shaft * d	S (min.)
0.3937	0.3941	9/16
0.4724	0.4728	21/32
0.5906	0.5910	25/32
0.6693	0.6697	7/8
0.7874	0.7878	1 1/32
0.9843	0.9847	1 1/4
1.1811	1.1816	1 1/2
1.3780	1.3785	1 3/4
1.5748	1.5753	

* Grind +0.0000 -0.0003

D Bearing	Housing D	H (max.)
1.1811	1.1817 to 1.1822	1
1.2598	1.2604 to 1.2609	1 1/16
1.3780	1.3786 to 1.3791	1 3/16
1.4567	1.4573 to 1.4578	1 1/4
1.5748	1.5754 to 1.5759	
1.6535	1.6541	

B Bearing	L Shaft	B Bearing	L Shaft
0.3543	19/64	1.2205	1 1/8
0.3937	21/64	1.2992	1 3/16
0.4331	3/8	1.3780	1 1/4
0.4724	13/32	1.4375	1 5/16
0.5118	7/16		

Special Centers With Standard Cutters

To the Editor:

● Books on machine design are well equipped with problems illustrating the formulas presented in the text. However, a good many problems in design work call for analysis requiring an understanding of fundamental conceptions rather than the substitution of data in the usual equations. To test out their powers of original thinking, seniors in machine design at Brown University are given problems, such as the following one, that require more than substitution in a formula:

In the present design of a machine two shafts are connected by gears having 36 and 159 teeth, respectively. The gears are 20-pitch, full-depth involutes with a 20-deg. pressure angle. Four of these machines can be sold provided the gear reduction between the shafts can be changed to a ratio of four to one.

The wall of the machine comes so close to the centerline of the shaft carrying the 36-tooth gear that the maximum outside radius of a gear on this shaft must not exceed 1 in. The wall cannot be cut out and the margin of profit is so small that the firm cannot afford to take the order if the shaft centers must be changed or idler shafts added. In others words, this order can be accepted only if the present drive can be replaced by two gears which can be generated with a standard hob or Fellows gear cutter on regular gear-cutting equipment.

Is there a practical solution for this gear drive?

—JAMES A. HALL
Brown University

Reversing Gear Mechanisms—I

ADAM FREDERICKS

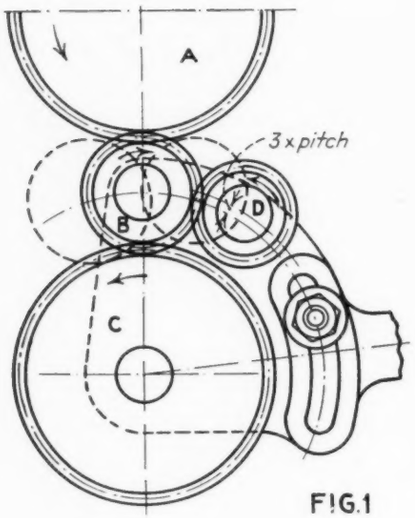
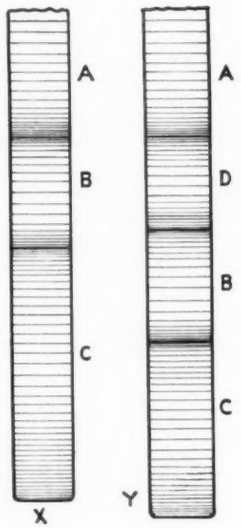


FIG.1



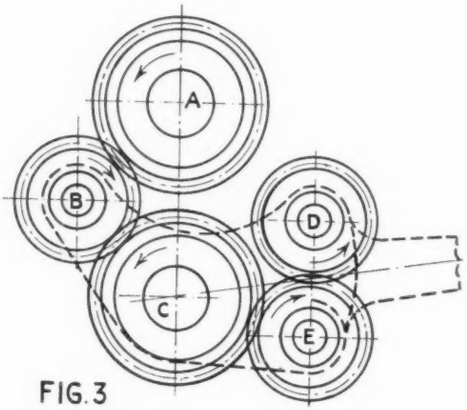


FIG.3

IN MANY kinds of machinery, reversal of direction is accomplished by means of gears. There are several forms of gearing from which to select, namely, the tumbler type of gear, bevel or spur gears with tooth or friction clutches, and planetary gearing.

When using a tumbler gear train for reversing direction of rotation, the clamping force on the swinging arm must be sufficiently great to withstand wedging of the gears when they run in one direction and the disengaging force when the gears run in the opposite direction. When an angular drive is required, bevel gears have an additional advantage of permitting a change of direction, by using a gear on each side of the center line of the central gear with one rotating and two stationary tooth clutches between them. Although being compact, bevel gearing is expensive because it requires an angular drive, making motor placement difficult. When tooth clutches are used for reversing, the power must be shut off when engagement takes place to avoid shock and noise.

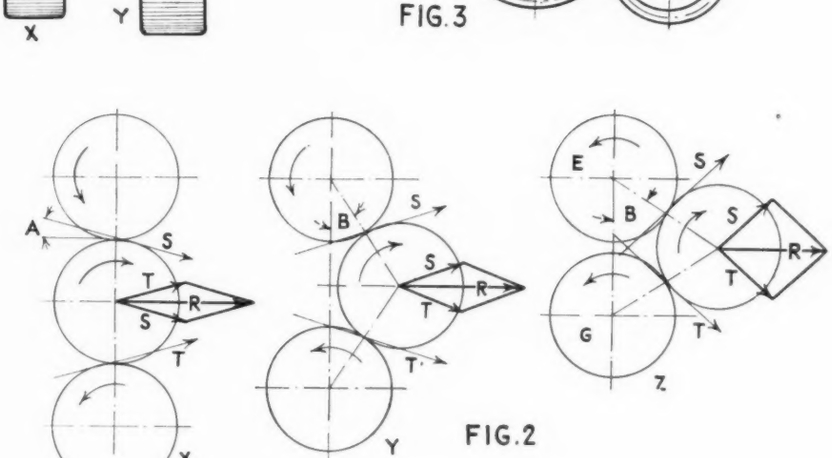


FIG.2

Fig. 1—Illustrates a tumbler reversing gear. Driven by gear *A*, forward drive is through pinion *B* to gear *C* shown at *X*. Pinion *D* meshing constantly with *B*, is made 3 teeth less than *B* and placed 3 tooth pitches above center line of *B*. Pinion *D* will clear *C* by 1 gear pitch. When rolled upward, *D* engages *A* while *B* disengages *A*. The drive will then be *A-D-B-C* as at *Y*. Dotted circles show the neutral position.

Fig. 2—At *X*, when three meshing gears are on the same center line the load on the center gear bearing is approximately double. The bearing load is *2 cos (tooth pressure angle)* times tangential load. This is proved by a parallelogram of forces at *X*, the resultant *R* being the load on the center bearing. If one gear is placed on a different center line, as shown at *Y*, the resultant bearing load *R* becomes less. At *Z* is the best possible method where the outside diameters of gears *E* and *G* just clear, 1/32 in. being sufficient. In both instances of *Y* and *Z* the resultant *R* can be calculated by the formula, *R* equals *2 cos (angle B-angle A)* $\times$ *tangential load.*

Fig. 3—A modification of Fig. 1. Although one more gear is used it is a better design as it incorporates the principle illustrated at *Z* in Fig. 2. This type of tumbler is locked in position by a spring-backed plunger in a knurled handle. Forward motion is through gears *A-B-C* and reverse is through gears *A-D-E-C*. Gear *A* is the driver. It should be so designed that the least number of gears are running for the direction used most, to save wear and eliminate additional backlash.

Fig. 4—A simple form of reverse gear is shown, operating as back gears of a lathe. In Figs. 1 and 3 the pinions are mounted on studs, overhanging the arm. At *Z* there is a bearing each side of the gear resulting in a sturdier construction. Only three gears are needed. At *X* the forward motion is illustrated by the arrow and the reverse at *Y*.

The throw *E* of the eccentric is regulated by the gear pitch. A 90 deg. movement of the handle at *Y* moves pinion *B* into engagement with driving gear *C*, while gear *D* drops from engagement. A 45 deg. movement places gears in neutral. Pinion *B* at all times engages gear *D*. The assembly may be locked by engaging a lock pin with the eccentric.

Fig. 5—A reverse drive is shown which may also be used as a hurry-up motion in one direction. Power comes through a sliding, keyed clutch *A* in the direction shown. When shifted to the right into the free running pinion *B*, double bevel gear *C* rotates toward the right. Shifting the clutch to the left drives gear *C* in the opposite direction through change gears *D* and pinion *E*. A one to one ratio is shown, but this can be altered by change gears *D* either increasing or decreasing the reverse driving speed. Only clutch *A* and gear *C* are keyed, other members running free.

Fig. 6—Although more compact than that in Fig. 5, it has not the flexibility in speed ranges. The drive may be transmitted through either shaft *A* or *B*. When shaft *B* is the driver and clutch *C* engages free pinion *D*, gear *E* revolves at a slow forward speed, and engaging clutch *C* with free gear *F*, gear *G* rotates at a faster reverse speed. Gear *E* is cupped out sufficiently to accommodate gear *F*.

Fig. 7—A reverse bevel gear with power coming through gear *A* which runs in direction of arrow. Pinions are mounted loosely on clutch shaft, the center or sliding clutch being keyed. When clutch is engaged on either side, the loose pinion clutch drives the sliding clutch and in turn the clutch shaft. The pinions are bronze bushed and held in place by collars or shoulders in the main casting. The surface speed of the bearing in the free pinion is double that of the pinion being driven because it is rotating in the opposite direction.

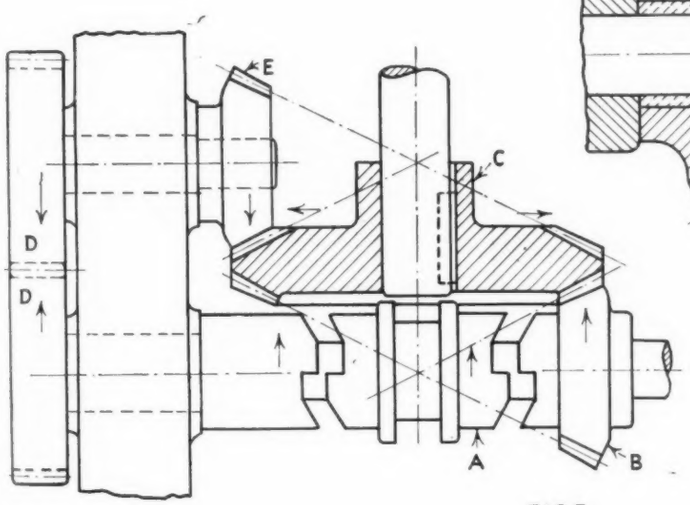


FIG.5

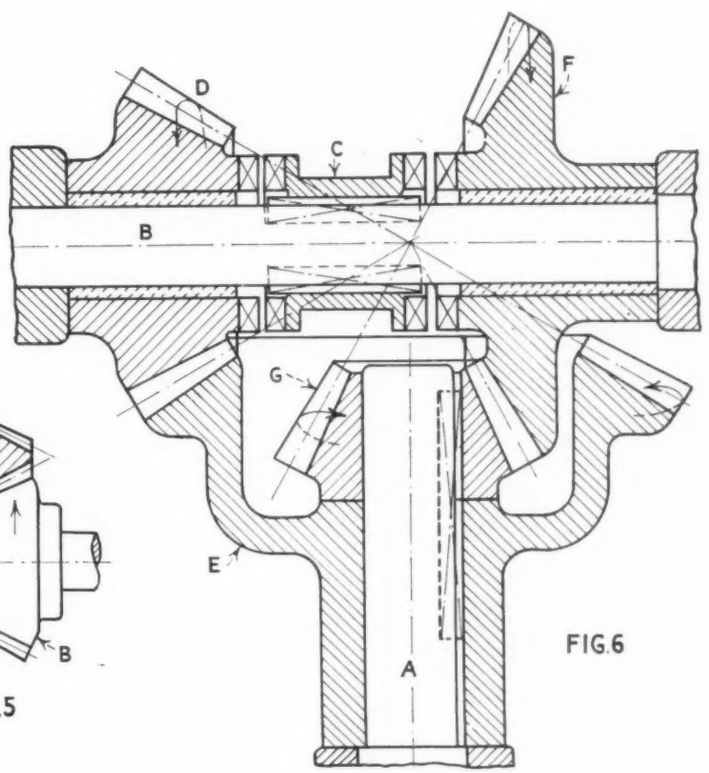
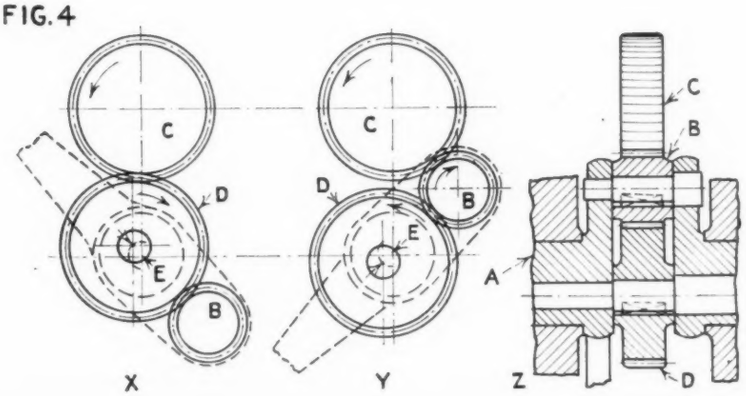


FIG.6

FIG.4



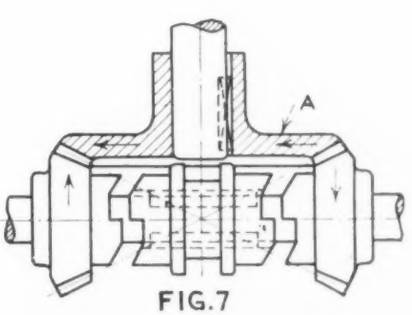


FIG.7

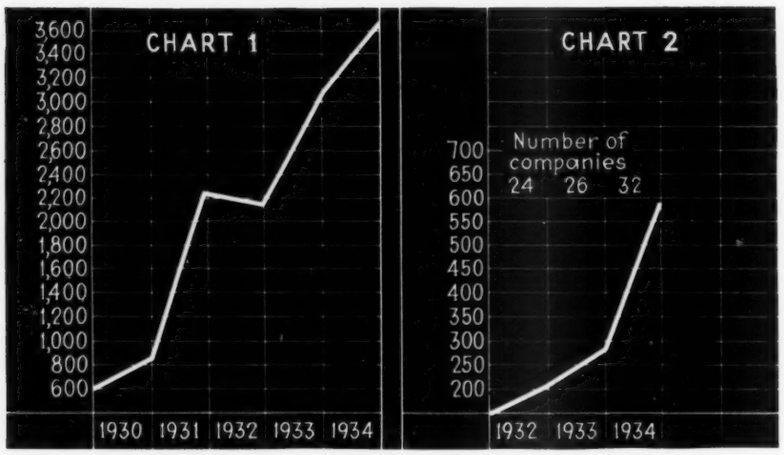


Chart 1—Increasing use of Parker Processes by thirteen major automobile manufacturers. Chart 2—Increasing use of Parker rust-proofing processes by makers of automobile accessories. Both Charts are to a scale based on units of rust-proofing energy

NEWS

More Rust-Proofing in Auto Industry

PRODUCTION records of the automobile industry forcefully demonstrate the increased appreciation of the value of phosphate coatings as rust-proofing agents. The accompanying charts, furnished by the Parker Rust-Proofing Company, reflect the increasing trend among automobile builders and accessory manufacturers to do everything possible to prolong the original fine appearance of their product as long as possible. In many applications, the rust-proofing assures extending the period of trouble-free mechanical performance.

Interviews with large users of rust-proofing have emphasized the cost-reducing factors inherent in the use of modern rust-proofing processes. Although there is a great saving in the fewer number of finishing coats required, there are also other economies of appreciable amount. One of these is the fewer number of rejects in the finishing department because perfect finish is much easier to obtain with rust-proofed sheets. This fact also reduces costs related to the inspection department as the inspection supervision in the finishing department need not be so close.

The accompanying Chart 1 shows the increased use of all types of Parker Processes used, according to a scale based on units of rust-proofing energy consumed. The increase is only partly explained by the number of customers (six in 1930, seven in 1931 and thirteen in the other three years). It is also the result of extending the use of the processes to a greater number of parts. Some who used Parkerizing only on mechanical parts, later started the use of Bonderizing on fenders and other sheet metal, following this with the use of Parcolite on bodies. While this chart covers the production of only thirteen major automobile lines, it should be understood that some part in practically every automobile built, or about 92 per cent of the total production, uses the Parker Processes to some extent.

Chart 2 covers the use of Bonderizing and Parkerizing by various accessory makers. These accessories include wheels, tire covers, lamp shells, horns, trunks, bumper parts, garage equipment, carburetors, locks, oil filters, license plates and many others. Although this chart is based on the same units of rust-proofing energy, in order to keep the charts comparable the scale was changed. It covers only the last three years, as earlier figures are not readily available.

ECPD Proposes Requirements for Professional Engineers

RECOGNIZED professional standing for engineers, calling for written and oral examinations, acceptable papers on engineering and cultural subjects, post-college study, and minimum experience qualifications was proposed by the Committee on Professional Recognition, one of the four divisions of the Engineers' Council on Professional Development, at a meeting in New York on March 7. Specific proposals for action by the cooperating organizations were also made by the other divisions of the ECPD including the Committees on Student Selection and Guidance, Engineering Schools and Professional Training.

These specific recommendations are the result of more than a year of study by the ECPD organization. As explained in detail in *Product Engineering* for November, 1933, page 432, the Council has only an advisory capacity, but its work is sponsored by five national engineering societies, as well as the Society for the Promotion of Engineering Education, and the National Council of State Boards of Engineering Examiners. Tentative proposals made by the ECPD have already been adopted by some of the sponsoring agencies, and the complete adoption of the entire program is expected as soon as the necessary action can be taken.

Progress reports and specific proposals for each of the four divisions were presented at the meeting. In reporting the activity of the Committee on Student Selection and Guidance, Dr. C. F. Hirshfeld, Chairman of the Council, announced that guidance literature and aptitude tests are now being used to select students for engineering courses. A 64-page booklet, "Engineering, A Career—A Culture," is being distributed by this committee to meet the need for information written for inexperienced students.

The Committee on Professional Training, with General R. I. Rees chairman, has prepared a 10-page printed booklet, "Suggestions to Junior Engineers," which includes a 5-page personal appraisal form, and a recommended reading list covering the fields of natural science, philosophy, economics, psychology, business, history, biography and fine arts. This committee is also promoting the organization of senior engineers throughout the country into groups to act as advisors and consultants to younger engineers.

In presenting the work of the Com-

mittee on Professional Recognition, Prof. J. W. Barker outlined the minimum qualifications originally set up by the committee (*P. E.*, November, 1933, page 433) and gave in detail the proposed program for certification. It is the expectation of the Council that this program in its final form will meet all requirements of the various engineering societies that now have varying standards for recognizing engineers and authorizing the use of the professional title.

Since the licensing of engineers has been adopted in 28 states and is under consideration in 17 others, Dean Barker held that one of the important functions of the ECPD would be to obtain uniform requirements. To assist in obtaining uniformity in future legislation, a model law is now being prepared.

Further consideration of the Committee's minimum definition of an engineer has led to a proposed program for progressive adjustment of qualifications during a transition period while the new standards are being established. According to this plan, qualification for certificates will include a "grandfather" clause decreasing in scope, and increasing requirements for applicants for examination during a 3-year transition period. It is proposed that the ECPD set up a Bureau of Certification with a staff to review applications, check references and credentials, and conduct examinations possibly as a joint agency with the National Bureau of Engineering Registration.

Written examinations for professional recognition, according to the recommendations of the Committee, would be met by submitting an original thesis or published professional paper covering an engineering project. including in the paper an analysis of the economic aspects of the project. Along with the thesis, the applicant would be required to submit a certified list of courses taken and books studied on cultural, civic and economic subjects together with an original essay or published paper on a similar subject. An oral examination would be given the applicant covering the field of these papers, and his continued post-college study and experience.

Heat-Treatment Cost in Selecting Alloys

THERE is an economic balance between the unit cost of the material and the cost of heat-treatment. Volume of producion, because of its effect upon the cost of heat-treatment, is often the principal factor in determining what alloy and what heat-treating process the manufacturer is to choose. In discussing this subject before the Chicago chapter of the American Society for Metals, John L. McCloud, chief of the chemical and metallurgical division of the Henry Ford Engineering Laboratory, stressed the possibility of attaining the most delicate control of heat-treatment at a unit cost no higher than that of much cruder treatment in small treating installations.

Speaking as a technician active in the outstanding mass production industry in the world, Mr. McCloud successfully demonstrated one extreme of material-treatment cost ratio. While he is known as one of the leaders in the introduction of alloy steels in American mass production, he emphasized that it would be entirely possible, technically, to make almost every part of a modern automobile of plain carbon steel, by giving close attention to the carbon content and subjecting the part to appropriate heat-treatment.

To the 500 members and guests who heard him, one of the most interesting and perhaps most astonishing features of Mr. McCloud's presentation was the series of lantern slides showing extremely complicated and expensive installations for heat-treating, often designed for a single part, which the Ford plant has been able to install.

An example was the machine for quenching camshafts. The operator took the shaft from the heating furnace and placed it in sockets in which it was rapidly rotated, while at the same time the shaft was carried past quenching jets. These jets were so arranged that the most drastic quenching took place only on the load-bearing portions of the shaft, for example, the cams and bearings.

MEETINGS

Society of Automotive Engineers—Summer meeting, The Greenbrier, White Sulphur Springs, Va., June 16-20. John A. C. Warner, secretary, 29 West 39th St., New York, N. Y.

American Society of Mechanical Engineers—Semi-annual meeting, Hotel Gibson, Cincinnati, Ohio, June 19-21. C. E. Davies, secretary, 29 West 39th St., New York, N. Y.

American Institute of Electrical Engineers—Summer meeting, Cornell University, Ithaca, N. Y., June 24-28. H. H. Henline, secretary, 33 West 39th St., New York, N. Y.

American Society for Testing Materials—Annual meeting, Book-Cadillac Hotel, Detroit, Mich., June 24-28. R. E. Hess, assistant secretary, 260 South Broad St., Philadelphia, Pa.

EXHIBITIONS

Industrial Arts Exposition — Rockefeller Forum, 30 Rockefeller Plaza, New York, N. Y., April 15-May 15. T. J. Maloney, manager, 386 Fourth Ave., New York, N. Y.

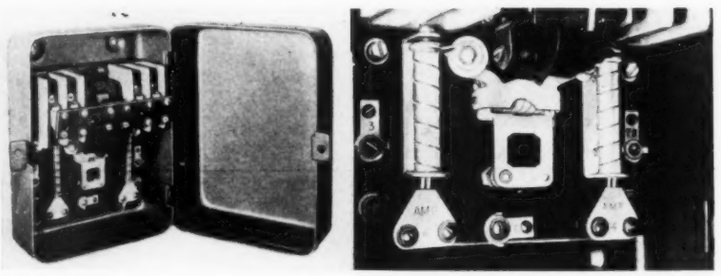
Electronic Control for 400 Hp. Motor

A NEW type of electric motor, which makes use of a "stationary commutator," has the characteristics of the series-type d.c. motor, but operates from an a.c. power source. This new development is made possible by recent advances in the application of electron tubes. Among the noteworthy features of this motor are the following: (1) While running from a.c. the motor has the characteristics of a series-type d.c. machine. (2) The speed of the motor is independent of the frequency of the power supplied. (3) Smooth control of the speed can be obtained over the full range. (4) In the event of a momentary interruption of the supply circuit, the motor will, upon the restoration of power, start and return to the speed at which it was previously operated, without drawing excessive current from the line. (5) The efficiency is high and the efficiency curve relatively flat which is a distinct advantage, particularly in the lower portion of the speed range.

Because of these characteristics, motors of this type are applicable to such auxiliaries as fans, centrifugal pumps, compressors, and similar equipment having load characteristics such that the series type of motor can be used to advantage. The motor has a stationary armature and a revolving field of the type used in synchronous motors. The armature, however, is provided with a special winding. Unidirectional current it supplied by means of a group of full-wave rectifiers which operate from the 3-phase, 60-cycle current source. The operation of the Thyratron rectifier is controlled by means of grids in these tubes so that power is supplied to the motor windings in the proper sequence and amount necessary to give the required torque for operation. This control is obtained by means of a small distributor mounted on one end of an integral distributor shaft. Speed control is obtained by varying the voltage applied to the motor armature, by means of a phase-shifting device, acting upon the same grids in the Thyratron tubes mounted on the control panel.

New Materials and Parts

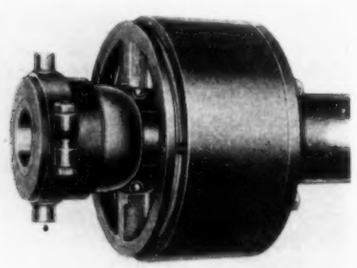
A.C. Motor Starter



These starters are intended for use with squirrel cage motors or as primary contactors for wound rotor motors in conjunction with suitable secondary rheostat. Standard features of design include: automatic reset overload; manual reset overload operative from button in cover; start and stop push button in cover; hand-off-automatic switch and reset button in cover, and fused or unfused disconnect switch. Thermal overload relay in the enlarged view, which is an integral part of all starters, is said to be so compensated so that wide variation in surrounding temperatures do not affect its operating characteristics. The relay rating can be changed by inserting proper heater elements to handle any load condition within the range of starter and, when determined, no further adjustments are necessary. Contacts are protected with adequate arc barriers which prevent flash-over between phases. Inclosures are of pressed steel for general purpose application, and cast iron for dust and water-tight applications. Monitor Controller Co., 51 South Gay St., Baltimore, Md.

Disk Clutch

Fully inclosed, single point adjustable clutch with single or double disks is suitable for use either as an integral part of the machine, or for general transmission of power. It is claimed that this clutch can run at high speed with transmitting capacity in direct ratio to speed. Friction disks are faced with "burn-proof" asbestos which permits extreme slippage in starting so a soft pick-up of load without vibration is possible. Working parts of shifting mechanism are heat-treated hardened steel. Shifter cone requires only a short movement to engage or release. Three sizes are available with capacities ranging from 1 to 6 hp. per 100 r.p.m. Made as a sleeve clutch with several types of sleeve bearings or as a cut-off coupling. The Kinney Mfg. Co., 3541 Washington St., Boston, Mass.



Fusible Caps

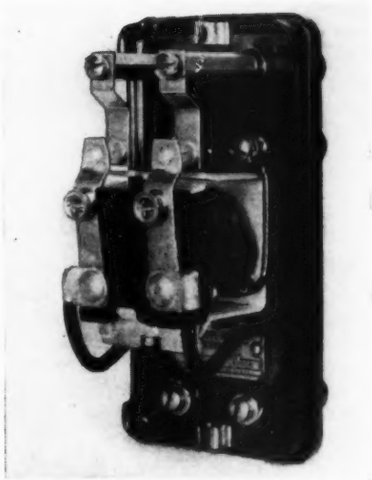
Blades machined from brass stock and dovetailed into supporting bakelite posts which prevent shifting out of alignment. The locking feature is said to be positive and prevents accidental breaking of current. Polarization is positive and prevents the insertion of caps in any but the right position. Grounding of the heavy ⅛ in. steel casing is positive. The grounding contact springs are of heavy phosphor bronze, clamped between bakelite and outer metal casing. The grounding blade is longer than the other, making contact first and breaking it last. This fusible cap is equipped with ½ in. thick sponge rubber disk, located in neck of steel casing to prevent entrance of dust and may be had with either adjustable metal cord grip or rubber cord grip bushings. Available either as 3-wire or 4-wire polarized caps. The 4-wire size is designed to take 3 standard 0-30 amp., 250 volt cartridge fuses and the 3-wire size will receive two standard 0-15 amp., 250 volt cartridge fuses. Harvey Hubbell, Inc., Bridgeport, Conn.

Koroseal

A plastic material, known as "Koroseal" is said to be superior to rubber for the manufacturer of a variety of products. While it is not expected to replace rubber in general use, it can be utilized in some instances where rubber consistency, combined with superior resistance to certain oils and chemicals and where flexing is required. It may be molded into any shape. The B. F. Goodrich Co., Akron, Ohio.

Auxiliary Relay

A small relay for use on a.c. or d.c. is suitable for switchboard or panel mountings. Type SK for switchboard mounting has a solid, molded cover, or molded cover with glass front; the front connected panel mounted relay uses no cover. Both types have two independent contact circuits, which may be adjusted for circuit-closing, or circuit-opening service. It is designed for continuous duty at rated voltage, and is obtainable for circuits up to 550 volts a.c., or 440 volts d.c. Ranges up



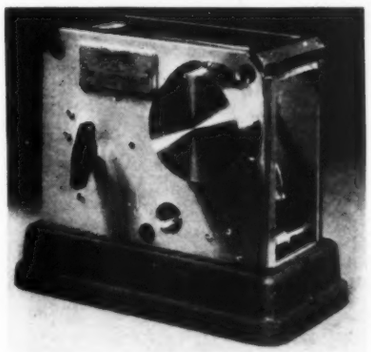
to 250 volts can be used for intermittent duties up to approximately three times their continuous value. The relay can be supplied with a coil for any current value up to 5 amp. The molded terminals are 1⅛ in. long. Each contact will carry 12 amp. continuous, and 20 amp. for one minute. Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

Convertible Motors

Single or multi-speed squirrel-cage and slip-ring motors, built to N.E.-M.A. standards are convertible to four types shown in photograph. Stator punchings are stacked between steel end rings and the entire assembly welded to the frame. By this construction, it is said that the stator core cannot shake loose. Rotor windings are assembled from round or rectangular hard drawn copper bars and are placed in the rotor slots without insulation or slot wedges. Available in all standard frequencies for service ranging from 110 to 220 volts, and from ½ hp., 600 r.p.m. to 125 hp., 3600 r.p.m. Harnischfeger Corp., West National Ave., Milwaukee, Wis.

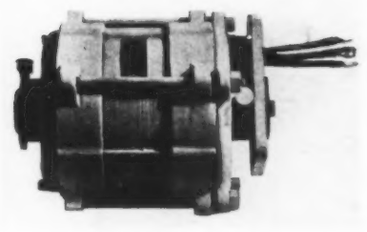
Selective Relay

Provides means of controlling the operation of one system when a definitely timed signal is transmitted over a second system in which the electromagnet of the relay is connected. The mechanism is spring driven, responding to approximately 60 operations of the controlling electro-magnet before it is run down. The mechanism is inclosed with bronze side plates and has beveled edge plate glass panels at the top and ends, which are set in grooves in the side plates. The relay is mounted upon a hollow iron base which incloses the controlling magnet and the terminal panel. The terminals are of the screw type. The base is finished with black crystal lacquer. Dimensions of this relay are 4⅜ in. by 7⅞ in. by 6⅛ in. high. Magnet resistance is 40 ohms, with a ½ amp. a.c. or d.c. contact capacity. The Gamewell Co., Newton, Mass.



Torque Motors

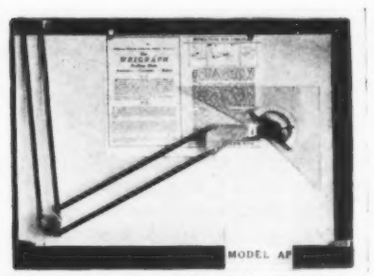
For individual installations where the duty cycle of each application is different, these motors are specially designed for services such as: reversing under load or plugging on the line, standing stalled on the line for long periods, alternate high and low speeds during a complete cycle, or



continuous and occasional operation. They may be used for controlling valves, operation of doors, power chucks, resetting of oil circuit breakers and other applications. Electric Specialty Co., Stamford, Conn.

Drafting Machine

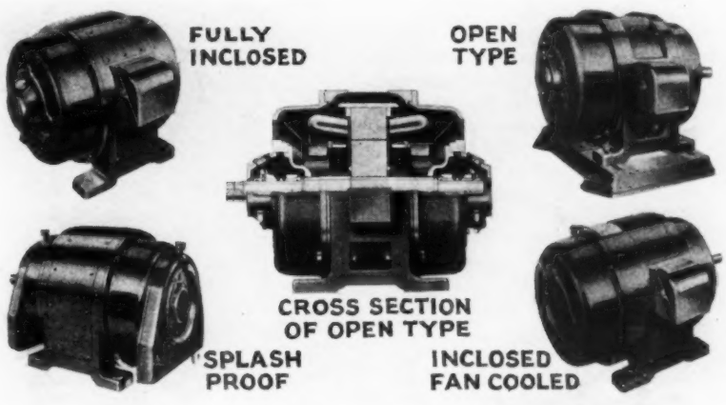
Light-weight drafting machine for rapid sketching for office or field use, which eliminates T-square and triangles. The device is mounted on a hard surface board. Paper clamps hold a single sheet or a whole pad of paper. The 30-45-60-90 deg. triangle is made of pyroxylin with ⅛ in. graduations on the two perpendicular sides. The small knob is loosened to turn the triangle in any direction. With one setting of the triangle, lines can be drawn parallel or perpendicular and 30 or 45 deg. to any other line. With one

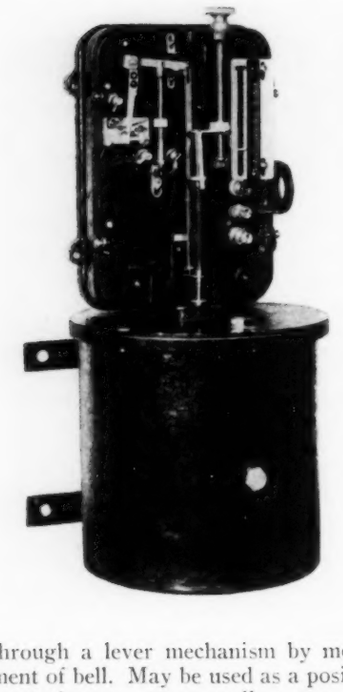


other setting, additional lines can be drawn in 15 deg. intervals. If small angles are required a 1 deg. protractor can be mounted on the triangle. The parallel device is made of nickel-plated metal parts, assembled with bronze bearings. Boards made in three sizes, 10x12 in., 12x18 in., and 18x24 in. L. G. Wright Co., 5713-9 Euclid Ave., Cleveland, Ohio.

Static Pressure Control

For air-conditioning and temperature control requiring accurate control of low pressures which normally do not exceed that of one inch of water above or below atmospheric pressure. Inverted bell with liquid seal houses a push-rod which extends upward through top of housing into head of control and operates control contact

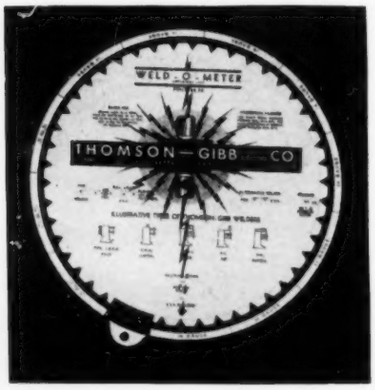




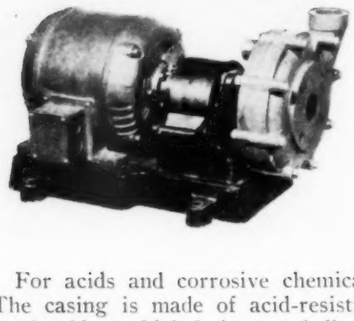
through a lever mechanism by movement of bell. May be used as a positive or static pressure controller, as a sub-atmospheric or draft controller, or as a differential controller. Interchangeability of pressure connection to accommodate the three types of control is quickly effected without any change to the static pressure controls proper. Minneapolis-Honeywell Regulator Co., Minneapolis, Minn.

"Weld-O-Meter"

A quick reference circular disk slide-rule chart that gives the type of welding machine required, the transformer capacity, the heat regulator setting and pressure needed to spot-weld metal sheets ranging from two thicknesses of 26 gage up to thicknesses of $\frac{1}{4}$ in. at production speeds of from 30 to 120 spots per min. for the lighter gages and 5 to 20 spots on the heavier stock. It also gives the current consumed per 1000 welds in kilowatt hours. Reverse side of chart tells similar information about the equipment needed to butt and flash weld metal ranging from 0.01 sq.in. to 2 sq.in. in cross-section area. Complete information is shown when the dial is set at one point on the chart which represents the area of the metal to be welded. This disk chart is being distributed to executives and engineers without charge. Thomson-Gibb Electric Welding Co., Lynn, Mass.



Non-Metallic Centrifugal Pump



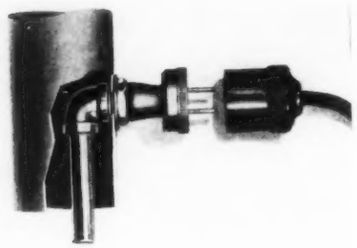
For acids and corrosive chemicals. The casing is made of acid-resisting hard rubber which is heat and distortion resisting. It is mounted to the base by means of an acid-resisting hard rubber covered casting which acts as a chamber to catch drip from the stuffingbox. By-pass is adjustable and cored into the back casing. The motor shaft is especially extended, tapered and pinned into the pump shaft with all parts interchangeable. This pump is available for medium capacities and up to 90 gal. per min., at a 12 ft. head. The American Hard Rubber Co., 11 Mercer St., New York, N. Y.

Irregular Curve

Automatic irregular curve which has an extruded rubber strip with a laminated core of Swedish spring steel. This improved device now has a drawing edge of thin metal strip fastened to the beveled outer edge of the rubber. The steel core places the drawing strip into an irregular curve different for every adjustment of the sliding clamp. The rubber strip prevents movement when the curve is being used. The instrument may be turned end for end to draw S curves. Available for drawing edge sizes from 7 to 60 in. The Wade Instrument Co., 2274 Brooklyn Station, Cleveland, Ohio.

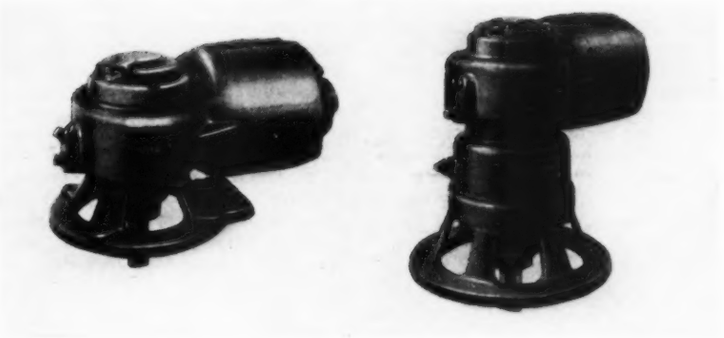
Electric Water Heaters

Installed in the lower hose connection between the radiator and engine of an automobile to heat the circulating water in cold weather. The unit is also available as a strip type heater, which is clamped to the intake manifold or engine block, or mounted on the inside bottom face of the radiator. Furnished with a rubber terminal plug with six feet of rubber covered extension cord for connection to a 115-volt outlet of the lighting circuit. Edwin L. Wiegand Co., 7500 Thomas Blvd., Pittsburgh, Pa.



Explosion-Proof Solenoid Switches

Type G is inclosed in a black enameled cast-iron case and is used in dry-cleaning establishments, gasoline filling stations, or wherever inflammable gases or vapors may be present. The heated gases created by accidental internal explosion, are cooled before reaching the atmosphere by the dissipation of their heat. Type B inclosed in a cadmium



plated, corrosion-resisting cast-iron case is made water tight by the use of live-rubber gasket between the cover and the base. The double-break silver alloy contacts require no maintenance and eliminate flexible connectors. There is no concealed wiring. Soldered-type thermal overload relays can be reset without opening the cabinet and are protected against overloads. Available for polyphase motors up to 15 hp., 110 volts; 30 hp., 220 volts, and 50 hp., 440-550 volts, and for self-starting and single-phase motors. Allen-Bradley Co., 1311 S. First St., Milwaukee, Wis.

Switch With Swing-Out Interior

The main feature of this switch is that the face and operating mechanism are mounted on a metal supporting plate which is hinged to the side of the box. By pulling on the operating lever, the entire interior

swings out and away from the box, leaving the entire box area open for unobstructed wiring and conduit connections. With the wire and conduit in place, it is merely necessary to swing the interior back into place and the connections can be made. Type D is available in not-fusible and plug and cartridge fusible, straight 2-pole, 2-wire solid neutral and 3-wire solid neutral switches, 125-250 volts and 250 volts, in 30 amp. sizes. Square D Co., Detroit, Mich.

Speed Reducers—Vertical

Type VMA shown at the left has a single reduction right angle worm drive. Worm is hardened and ground steel and worm wheel is of phosphor bronze. The reducers in both illustrations are mounted on anti-friction bearings, and have oil seals to prevent leakage. Type VMA has torque capacities up to 263 lb.-in., for continuous running at its rated capacity. Available in ratios of $3\frac{3}{4}$ to 1 up to 48 to 1 and with $\frac{1}{4}$ or $\frac{1}{3}$ hp. a.c. or d.c. splash-proof motor. Can be furnished without motor.

Type CVMA at the right has a compound reduction, right angle gear drive. The first reduction is by worm wheel and the second reduction through a series of spur gears. The shaft may extend either side for two widely different speeds. Has torque capacities up to 1500 lb.-in., for continuous running at its rated capacity. Available in ratios up to 5,000 to 1 with or without mounting base and with $\frac{1}{4}$ hp. a.c. or d.c. splash-proof motor. Can be furnished without motor. Boston Gear Works, Inc., North Quincy, Mass.

Drafting Table With Sliding Chair

Made of a combination of steel and wood, the "Steelwood" table has a height of 37 in. instead of the standard 30-in. height, which eliminates the

use of raising blocks. The drawing board is made of sugar pine. Cadmium-plated steel end cleats are attached to accommodate the parallel rule device and provide accurate surfaces for the T-square. The board is adjustable for slant with a tilting device on each side. Front legs of chair are mounted on a lateral track attached to the table with ball-bearing fiber wheels running in the channel guide. Outside legs have ball-bearing casters which roll on the floor. Chair is said to provide an effortless lateral movement in the front of board. Seat is pivoted at the lower part of frame to permit forward or backward movement, and can be adjusted for height and proper distance from table to fit individual needs. A battery of three lights is available which can be slid as a unit in a channel at upper edge of board to give illumination where needed. Hamilton Mfg. Co., Two Rivers, Wis.

MANUFACTURERS' PUBLICATIONS

Ampco Metal—Ampco Metal, Inc., 3830 West Burnham St., Milwaukee, Wis. Thirteen engineering sheets, 8½x11 in., of which a new series will be published monthly. Gives properties and characteristics of a metal composed of copper, high aluminum, and high iron alloy. Describes application, corrosion resistance to various chemicals, metals for gears and machining data and illustrates various grades with micro-photographs.

Automatic Timers—Walser Automatic Timer Co., Chrysler Bldg., New York, N. Y. Catalog, 4 pages, 5½x8½ in., illustrating various types of industrial timers and describing functions, construction and sizes. Application to numerous classes of equipment listed.

Bakelite Varnish, Enamel, Lacquer and Cement—Heat-Hardenable—Bakelite Corp., 247 Park Ave., New York, N. Y., booklet 125-A, 39 pages, 6x9 in., describing the properties, uses and methods of application of synthetic resins which require baking to bring out their best properties.

Bearing Bronze Bars — Johnson Bronze Co., New Castle, Pa. Bulletin 340, 32 pages, 4½x11 in. A stock list of bushings, bearings and machined bar bronze, made of copper 80, tin 10 and lead 10. Inside and outside dimensions and weights of 182 sizes of cored bars and 33 sizes of solid bars are given. Flanged type, bronze graphited and electric motor bushings are listed as well as styles of grease and oil grooves.

Beryllium Copper—The American Brass Co., Waterbury, Conn. Publication B-21, 19 pages, 8½x11 in. Illustrates application and describes composition, fabricating procedure, physical and mechanical properties and includes wear charts. Free Cutting Brass Rods—Publication B-14, 19 pages, 8½x11 in. Describes free cutting brass, bronze, leaded copper and nickel silver rods, with comparative production figures of brass and other metals, with tables of weights of round and hexagon brass bars.

Chains—Morse Chain Co., Ithaca, N. Y. "Chain Length Calculator," 5x9 in., is made of cardboard with an adjustable slide for direct reading of chain lengths required (number of links) for any combination of sprockets, any pitch and any center distance listed, for silent and roller chain drives. Knowing the center distance, the number of teeth in each sprocket, the chain pitch and the number of links in the chain may be read at one setting of the slide.

Duprene—E. I. du Pont de Nemours & Co., Wilmington, Del. Small 11-page illustrated booklet, "A story of Man-Made Rubber," describes composition and properties and lists numerous applications to various industries.

"Kunial" Brass—I.C.I. Metals, Inc., Kynoch Works, Witton, Birmingham 6, England. Pamphlet, 6x9¼ in., 25 pages, describing a brass of copper, zinc, nickel and aluminum alloy. Mechanical properties, temperature charts, photo-micrographs and applications are included.

Leather Packings—Alexander Brothers, Inc., 406 N. Third St., Philadelphia, Pa. Booklet No. A-49, 24 pages, 3½x6¼ in., shows and describes a number of leather packings of various types, including cup shape, "U," flange, Vee, flat and hydrant valve leathers for hydraulic and pneumatic equipment.

Manganese Steel—The American Manganese Steel Co., Chicago Heights, Ill. "The Amsco Bulletin" for January, 1935, gives a story of manganese steel including history, chemical content, physical properties, method of manufacture, heat-treatment, machinability and welding. Test pieces are shown with charts and photo-micrographs, as well as industrial applications.

Motorized Speed Reducers—Link-Belt Co., 2045 W. Hunting Park Ave., Philadelphia, Pa. Book No. 1515, 20 pages, 8½x11 in. Describes flange-mounted motor gear reducers with double and triple reduction, using helical gears of heat-treated alloy steel, in ratios of 38.5 to 1, and 292 to 1, and ½ to 75 hp. and ½ to 30 hp., respectively. Illustrated with photographs and dimension diagrams.

Paint—Sherwin-Williams Co., 101 Prospect Ave., N.W., Cleveland, Ohio. Booklet "Guide to Profit Through Paint," 34 pages, 8½x11 in., describes plant conditioning, as "putting your industrial house in order," and "plant conditioning is plant painting, its first and last objective being to lower unit costs of your products—be they textile, metal, food, process, chemical, plastics, paper or other lines." Protection, appearance, light and sight, color and unit costs are discussed.

Relays and Timing Devices — Struthers Dunn, Inc., 139 N. Juniper St., Philadelphia, Pa. Folder giving a short description with illustrations of their entire line of relays, timing devices, electric counters, thermostats, electric pots and ladles.

Stainless Steel—Republic Steel Corporation, Massillon, Ohio. Booklet, 16 pages, 8½x11 in., contains information on physical properties at high temperatures and the forging, flanging, annealing, deep drawing, punching and shearing, riveting, welding, soldering, brazing and machining of various types of 18-8 steels. Numerous photographs and applications are given. A table of the degrees of corrosion resistance to several hundred chemicals, solutions and other reagents is included.

BOOKS AND BULLETINS

Torsional Vibration

W. A. Tuplin. 320 pages, 5½x8½ in. 89 illustrations. Red clothboard covers. Published by John Wiley & Sons, Inc., 440 Fourth Ave., New York, N. Y. Price $5.50.

Elementary theory and design calculations are presented for practical engineering problems in rotating machinery, with particular attention given to internal combustion engines.

Fundamentals necessary for an understanding of the problem are included in earlier chapters on systems of one or more masses and branched systems. But the design point of view is not obscured by the mathematical and analytical presentation. Detailed examples of actual calculations are given a prominent part in each chapter.

The author carefully explains the more difficult mathematical steps required in the solution of vibration problems. Separate chapters are devoted to methods of solving numerical equations and the analysis of harmonic vibrations.

Forced vibrations, critical speeds, vibration dampers, torque fluctuations and torsional impact are each treated in separate chapters, making extensive use of examples and sample calculations. Since it is written specifically for the designer, this text offers practical assistance in everyday torsional vibration problems.

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Concerning Graphoid Surfaces and the Parts they Play in Lubrication

Acheson Colloids Corp., Port Huron, Mich. Technical Bulletin No. R-220, 5 pages.

A review of the benefit of adding colloidal graphite to oils for the purpose of forming graphite surfaces on friction parts of mechanical devices. It is stated that "modern research has not only confirmed the existence of graphoid surfaces, but has also revealed facts which serve to explain why the addition of colloidal graphite to petroleum bodies lowers their original coefficient of friction values, raises their critical temperature and acts as an effectual adjunct lubricant at high temperatures under boundary conditions." The question "what is oiliness?" is discussed and the fundamental facts about lubrication relative to thin (partial) film and thick (full) film are explained.

Photo-Elastic Properties of Soft Vulcanized Rubber

National Bureau of Standards, Research Paper RP 751, by Wilfred E. Thibodeau and Archibald T. McPherson, 10 pages, 5 cents.

A study of photo-elastic properties of transparent soft vulcanized rubber, used as samples for investigation of stresses in models by the photo-elastic method. Three typical rubber compositions were examined and values of the coefficient of retardation and the stress-optical coefficient were found when subjected to relatively wide ranges of stress. The results of these tests are expressed in simple equations for each compound. The constants of these equations, in turn, were related to the composition of the rubber.

Industrial Electronic Tubes

Westinghouse Electric & Manufacturing Co., East Pittsburgh, Pa. Pamphlet, 217 pages, $2.25.

In this zincograph pamphlet the treatment is non-mathematical, starting with electronic phenomena of the mercury arc, passing to high-vacuum tubes and the grid-glow family, thence to photo-responsive electronic devices and cathode-ray tubes and ending with the specific technique of a variety of applications. Through the introduction of this electronic phenomena and the circuits built around them, it is the purpose of this book to show how the physical properties can be put to practical use in industrial manipulation and control.

As a supplement to the above, a "Manual of Experiments" may also be purchased for 65 cents, which tells of a series of 24 tests, the apparatus needed, method of set-up, procedure to be followed and the results to be obtained

Chart for Unit and Total Weight and Number of Pieces

CARL P. NACHOD
Vice-President
Nachod & United States Signal Company

TABLES to give the number of pieces of given unit weight required to make a certain total weight would be too large and cumbersome. By the use of the nomogram shown on the following page, a complete range of values can be included in a single chart. Although the accuracy of the nomogram reproduced here is limited because of its size, the chart can be constructed to any desired scale by the method described.

Three equally spaced parallel lines are drawn perpendicular to a base line. These lines are graduated, using a logarithmic scale. Line *A*, on the left, and line *C* on the right are graduated from 1 to 10 and a reciprocal scale is added to line *A*. The center or index *B* is graduated from 1 to 100. These scales can all be transferred from a conventional slide rule.

A single line drawn through points on two of the lines gives the remaining factor on the third line. Thus, for 675 (6.75) pieces per lb. on scale *A*, and 128,000 (12.8) pieces on scale *B*, the dotted line cuts scale *C* at 1.9 or 190 lb. total weight.

The reciprocal scale also solves problems involving weight per piece. For example if the weight per piece including waste is 1.48 lb. (scale *A*), stock weighing 190 lb. (scale *C*), will make 128 pieces (scale *B*). The decimal point can usually be determined by inspection.

The same scale divisions can also be used to find the weight for a given volume of various materials. For this problem scale *A* designates volume in cu.in., and scale *B* gives total weight. A line drawn from the reference point for aluminum on scale *C* to 4 cu.in. on the volume scale, indicates on the center scale a total weight of 0.37 lb., the weight of 4 cu.in. of aluminum.

Several other problems can be solved by this chart. The product of any two numbers on the outer scales is given on the middle scale. A horizontal line connecting the same number on the outer scales cuts the middle scale at the square of the number; similarly, a horizontal through any number on the inner scale cuts the square root of the number on the outer scales. In addition, the numbers opposite each other on scale *A* are reciprocals.

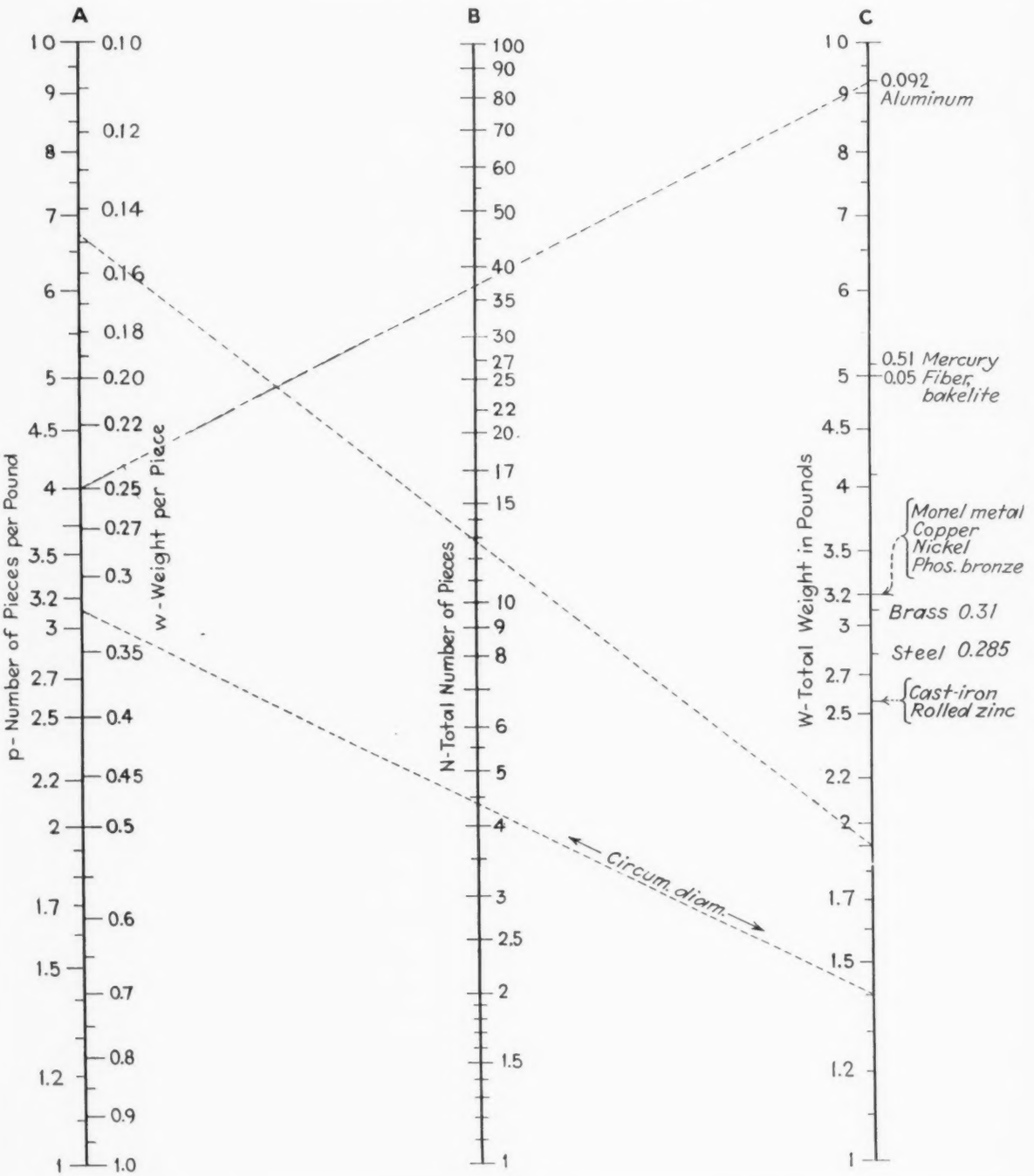
When the diameter is given, the circumference of a circle can also be obtained. A line from the point 3.14 on scale *A* intersects the diameter on scale *C*, and the circumference on the center scale. Thus, a circle having a diameter of 1.4 (scale *C*), has a circumference of 4.4 (scale *B*), as shown by the dotted line.

To find the circumference of a circle when the diameter is given, lay a piece of tracing cloth or waxed paper over the chart. Trace the line *B*, and draw lines from the point 7.85 on scale *B* to the points 1 and 10 on the *C* scale. When this k-shaped figure is moved up and down along the line *B*, the intersection on *B* is the area of the circle having for its diameter the value cut on *C* by either arm of the k-shaped figure. By using the point 8.66 on scale *B* in constructing the auxiliary figure, the same procedure gives the area of a hexagonal cross section when the distance across flats is set on the *C* scale.

PRODUCT ENGINEERING © REFERENCE BOOK SHEET

Chart for Unit and Total Weights

CARL P. NACHOD
Vice-President
Nachod & United States Signal Company



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